PRINCIPLES OF MECHANICS

BY

W. J. MILLAR, C.E.

PRINCIPLES OF MECHANICS,

AND THEIR APPLICATION TO

PRIME MOVERS, NAVAL ARCHITECTURE, IRON BRIDGES, WATER SUPPLY, &c.

THERMODYNAMICS, WITH SPECIAL REFERENCE TO THE STEAM ENGINE.

BEING AN ABSTRACT OF LECTURES

DELIVERED TO

THE CLASS OF CIVIL ENGINEERING AND MECHANICS IN THE UNIVERSITY OF GLASGOW, SESSION 1872-73.

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PREFACE.

As indicated on the title-page, the subjects treated of in this book constituted in a more extended form a series of Lectures delivered to the Class of Civil Engineering and Mechanics in the University of Glasgow during the latter part of session 1872–73.

Shortly after the death of Professor Rankine, the author was appointed to conduct the class referred to during the Professorial vacancy; and the various subjects treated of formed part of the complete course as entered in the syllabus of the class.

It having occurred to the author that a carefully-revised abstract of these Lectures might be of use to students and others studying the various subjects treated of, the work as contained in the following pages is the result.

The subjects have been treated of as concisely as possible, numerical illustrations being occasionally given to assist the reader.

Various authorities have been consulted in the preparation of the present work; amongst others,

Professor Rankine's Works;
Moseley's Engineering and Architecture;
Fairbairn's Mills and Millwork;

Deschanel's Natural Philosophy, by Prof. Everett;
Shipbuilding in Iron and Steel (Reed);
Transactions Inst. Civil Engineers;
Transactions Inst. Engineers and Shipbuilders in Scotland;
Transactions Inst. Naval Architects;
Report (British Assoc.) Sea-going Qualities of Ships, 1869;
Annual of the Royal School of Naval Architecture and Marine Engineering;

and the various Engineering and Scientific periodicals, &c.

W. J. M.

GLASGOW, *October*, 1874.

CONTENTS.

PRINCIPLES OF MECHANICS.

ENERGY.

THE meaning of the term energy is capacity for performing work, or of overcoming resistance. This term has been further distinguished as follows:—

1. Potential energy, which means a capacity or ability to perform work in virtue of position.

Example.—A weight of 10 lbs. placed at a height of 10 feet above the ground, is said to have 10×10 foot pounds of potential energy.

2. Actual energy, or the capacity of performing work in virtue of motion against a resistance. This term applies to moving bodies, and is expressed by the symbol $\frac{W v^2}{2 g}$, where W represents the weight of the body in lbs., v the velocity of feet per second, and g equal to the force of gravity at the earth's surface; or generally, $g = 32 \cdot 2$.

Example.—A body weighing 10 lbs. is moving with a velocity of 10 feet per second; then the actual energy of the body is $= \frac{10 \times 10^2}{2 \times 32 \cdot 2} = 15 \cdot 5$ foot pounds. The symbol $\frac{v^2}{2 g}$ represents the height through which a body would fall to acquire the velocity v.

The general expression of the law of energy is, that

where energy is exerted work is performed, and may be expressed by symbols as follows:—

$$P \times s = R \times s_1, \text{ or } \int P\,ds = \int R\,ds_1,$$

where P represents the effort applied, and s the space through which the effort moves; R the resistance overcome, and s_1 the space through which the resistance is driven. The expression $\int P\,ds = \int R\,ds_1$ means that the total energy exerted is equal to the total work performed, and may be represented by a diagram; the symbol $\int$ indicating the summation of the term to which it is attached.

The total resistance overcome is made up of the useful work done, the resistances such as friction, and the resistance due to the mechanism of the machine.

Example.—If a weight of 100 lbs. be raised through 10 feet by means of a simple combination of cord and pulley, $P \times s = \Sigma R \times s_1$, where $\Sigma R \times s_1$ represents the sum of the resistances, viz. the weight to be raised multiplied by the height through which it is raised, and the friction of the pulley multiplied by the distance through which that friction is overcome.

Work is measured by foot pounds, a foot pound being one pound raised through a distance of 1 foot; and if we express the *rate* at which this work is performed, we have one pound raised through 1 foot in one second. Thus 550 lbs. raised through 1 foot in one second, is what is called one horse-power, or 550 foot pounds of work performed per second.

Work performed then represents energy exerted, or *power*.

Energy is transmitted through the medium of machines.

The *power* of a machine is the measure of the energy exerted,

And the term horse-power is the rate at which a given quantity of work is performed.

PRIME MOVERS.

Prime movers are machines or appliances whereby we are enabled to utilize energy. These may be classed as follows:—1st, Animals; 2nd, Water Wheels and Water Engines; 3rd, Windmills; 4th, Steam, Heat, and Gas Engines; 5th, Electro-magnetic Engines.

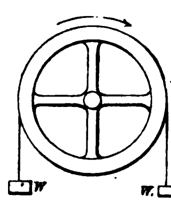
In these machines energy is said to be exerted through various forms of *power*, as 1st, Muscular Power; 2nd, Water Power; 3rd, Wind Power; 4th, Heat Power; 5th, Electric or Magnetic Power.

DYNAMOMETERS.

Dynamometers are instruments for measuring work, and they generally consist of an arrangement whereby a resistance such as friction is overcome, through a given distance in a given time, the rate of motion being uniform.

If the wheel A have a cord passing half round its circumference, and weights W and W_1 be applied at the ends of the cord; then if the wheel be at rest, these weights must be equal; but if we now turn the wheel in the direction indicated by the arrow, we shall find that in order to keep the cord in position, so that it shall not travel round with the wheel, the weights will no longer be equal, but W will exceed W_1, and the difference of these weights will be the frictional resistance,

FIG. 1.



which being multiplied by the velocity of the circumference of the wheel, will give the work done.

Example.—A wheel over which passes a cord having weights W and W_1 hung at its ends (W being = 100 lbs. and $W_1 = 14$ lbs.), is found to make $1\frac{1}{2}$ revolution per second. The circumference of the wheel is 3 feet. The resistance in this example is $W - W_1 = 100 - 14 = 86$ lbs., and the space through which this resistance is overcome is $1\frac{1}{2} \times 3 = 4\frac{1}{2}$ feet per second. The work done then $= 86 \times 4\frac{1}{2} = 387$ foot pounds per second, or $\frac{387}{550} =$ $\cdot 7$ horse-power. In such cases, other things being equal, the ratios of the tensions or pulls at the ends of the cord vary as the power of the number of turns; thus, ratio of tensions with 1 turn $= 100^1 : 1 \cdot 96^1$, then ratio of tensions with $\frac{1}{2}$ turn $= 100^{\frac{1}{2}} : 1 \cdot 96^{\frac{1}{2}}$, or as $100 : 14$; and generally if R = ratio of tensions with 1 turn, then R^n = ratio of tensions with n turns.

Professor Sir Wm. Thomson proposes to use this arrangement as a dynamometer, by passing the cord round the fly-wheel of an engine.

There are various forms of dynamometers, such as Morin's, White's, and Prony's.

MUSCULAR POWER.

We may consider the origin of this power to be due to chemical combination, through organic processes, such as in living animals.

Animals perform work in various ways, such as by dragging loads or raising weights. The horse exerts his power to most advantage through traction, as in drawing carts, giving on an average about $12\frac{1}{2}$ million foot pounds per day, and that when the resistance to traction is equal

to 120 lbs. drawn at a rate of about $3\frac{1}{2}$ feet per second during eight hours. A man exerts his power to the most advantage by raising his own weight, as in ascending heights, giving as an average upwards of 2 million foot pounds per day; the weight being 143 lbs. raised through a vertical height of $\frac{1}{2}$ foot per second during eight hours. When rowing, a man performs about $1\frac{1}{2}$ million foot pounds.

WATER POWER.

The origin of water power is solar heat through evaporation, from the surfaces of oceans and lakes, and subsequent condensation. The water which at one time occupied a certain position on the earth's surface, is now deposited at a higher level, and this water by suitable means may be utilized for the performance of work.

STORAGE OF WATER.

Water is either stored naturally or artificially; in the former case by means of natural rock basins, such as lakes; and in the latter case by means of artificial lakes, generally called dams or reservoirs. The natural outlet to a lake is a stream or river, and by this means the water may be conveyed for the purposes of obtaining power; where a dam or reservoir is formed, a channel must be constructed to serve the purposes of the river. In selecting ground for storage purposes, the rainfall of the district must be considered; this rainfall is very various, and depends upon the situation and configuration of the ground. In this country the winds most laden with moisture are from the west and south-west, and consequently the rainfall is greatest in districts lying on the west coast, such as the lake district in England;

the western coast of Ireland; and the islands and western coast of Scotland. In mountainous districts the rainfall is greater than in flat districts; the reason of this is, that the moisture-laden currents of air are deflected upwards by the mountain slopes, and by the subsequent expansion due to decrease of pressure, they fall in temperature, and condensation ensues. The cold surfaces of the mountains also assist in this condensation.

The rainfall is measured by instruments called rain gauges, which are simply cylindrical vessels into which the rain falls, and which is measured at stated intervals, either by means of pouring out the contained water into a graduated measuring vessel, or by having a floating graduated scale in the gauge vessel itself.

An artificial lake or reservoir is generally formed by running an embankment across the lower part of a valley, through which a stream flows, and the waters collected constitute a store of potential energy, which can be changed into actual energy by allowing part to flow off through suitable channels. These embankments are usually constructed of earth carefully deposited in layers; and in order to render this earth wall impervious to water, a wall of clay in the state termed *puddle* is built up along with, and in the centre of the earthwork. The stability and proportions of embankments and walls are considered under "Stability of Structures."

The water having been conveyed to the *site* where the machinery for transforming the energy of the water into useful work has been placed, we have now to consider by what means this energy may suitably exert itself: we thus come to consider the various forms of prime movers for the development of water power.

WATER WHEELS.

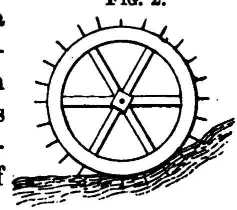
This class may be subdivided into 1st, Vertical; and 2nd, Horizontal Water Wheels. Under the 1st division we have Undershot Wheels, Overshot Wheels, and Breast Wheels; under the 2nd class we have Turbines.

I.—*Vertical Water Wheels.*

The original or elementary form of water wheel was the undershot. This wheel, which was of wood, consisted simply of an axle from which arms radiated, the extremities of which were connected by boards shaped to the curve, and to which float-boards were attached. The float-boards were flat, and were placed radially, as shown in Fig. 2.

These wheels were sometimes placed between boats moored in a stream or tide-way. The power required in connection with the London Water Works in Smeaton's time was drawn from large wheels of this description placed between the piers of Old London Bridge.

FIG. 2.



Iron is now generally used in the construction of water wheels. In the undershot wheel the water acts usually by impulse, sometimes by weight.

Relation of the Terms Impulse, Momentum, and Actual Energy.

I. Impulse is defined to be the product of a force into the time during which the action continues or that the effort is exerted. If F represent the force and t represent the time, then $F\,t$ = the impulse.

II. Momentum is defined to be the product of the

mass of a body into the velocity with which the body is moving, and is expressed by $\frac{W v}{g}$. If we consider this formula, we shall see that since $\frac{v}{g} = t$, in a falling body, the momentum of the latter is simply the weight multiplied by the time, or $W t$; and this term becomes a measure of the importance of the force exerted, and thus resembles a moment of the value $W \times L$, where $W =$ weight and L equal the length of the arm at which W acts.

III. Actual energy or accumulated work is represented by $\frac{W v^2}{2 g}$. Now to show the relation between impulse, momentum, and actual energy, we have

1. Impulse $= F t$.

2. Momentum $= \frac{W v}{g} = W t$, in a falling body.

3. Actual energy $= \frac{W v}{g} \times \frac{v}{2} = \frac{W v^2}{2 g}$, $\frac{v}{2}$ being the mean velocity.

1. *Undershot Wheels, in which the Water acts by Impulse.*

In these wheels the float-boards are radial and are plane surfaces, on which the water impinges, and by its pressure produces motion. This pressure is due to the flow of water upon the boards.

There are three cases to be considered:

1st. When the board does not move.

2nd. When the board moves at the same rate as the water.

3rd. When the velocity of the board is less than that of the water.

In the 1st case we have a part, or the whole of the

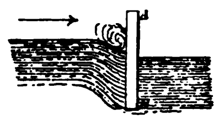
energy of the water spent by the water striking the board.

In the 2nd case, the board offers no resistance to the water, and therefore no energy is expended.

In the 3rd case, a certain amount of energy is expended in overcoming the resistance of the board through a certain space.

Let the float-board A B be driven forward by the action of a current of water, and if the velocity of this current of water before striking the board be V, and the velocity of the same current after striking the board be v, then the expression $V - v$ will represent the velocity lost by the water during its action upon the board.

Fig. 3.



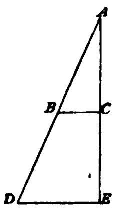
The board then has opposed a certain resistance to the energy of the water, and in order to see how much energy has been expended in overcoming this resistance, we have $\frac{V - v}{g} \times \frac{V + v}{2} \times W = \frac{W (V^2 - v^2)}{2g}$, that is to say that if we find the total energy of the supply current or $\frac{W V^2}{2g}$ and from that deduct the energy still left in the water or $\frac{W v^2}{2g}$, or $\frac{W V^2}{2g} - \frac{W v^2}{2g} = \frac{W (V^2 - v^2)}{2g}$, we have the energy spent upon the board expressed by the formula $\frac{W (V^2 - v^2)}{2g}$, where $W =$ weight of water per second which strikes the board, and $g = 32 \cdot 2$.

To explain the formula already obtained we may consider the case of a falling body. Let a body of weight W fall from a point; its velocity at the end of one second will be $32 \cdot 2$, and at the end of two seconds its velocity will

be 64·4, and so on, adding equal increments of velocity in equal increments of time, that is because the accelerating force of gravity or 32·2 is taken as being uniform.

Let A be the point at which the body starts, and let $\overline{BC}$ represent its velocity at end of one second $= v = 32 \cdot 2$, and let $\overline{DE}$ equal velocity at end of two seconds $= V = 64 \cdot 4$, then will A D E be a triangle, and it is evident that the total energy of the body when passing the point E will be $= W \times \frac{V}{2} \times \frac{V}{g} = \frac{W V^2}{2g}$, because $\frac{V}{2} =$ mean velocity during the time $\frac{V}{g}$. Again, the energy of the body when passing the point C will be represented by $W \times \frac{v}{2} \times \frac{v}{g} = \frac{W v^2}{2g}$, so that the energy *gained* between C and E is equal to the difference of these energies, or $\frac{W V^2}{2g} - \frac{W v^2}{2g} = \frac{W (V^2 - v^2)}{2g}$; or we may express the energy gained between C and E as follows: $W \times \frac{V - v}{g} \times \frac{V + v}{2} = \frac{W (V^2 - v^2)}{2g}$, because $\frac{V - v}{g} =$ time occupied between C and E, and $\frac{V + v}{2} =$ mean velocity during that time. It is also evident that since $\frac{W (V^2 - v^2)}{2g}$ represents the energy *gained* when the speed of a body is increasing, it will represent the energy *lost* when the speed of the same body is decreasing in the same ratio; the latter application concerns the problem of the float-board, and explains the formula already given for that case.

FIG. 4.



2. *Undershot Wheels, in which the Water acts by Weight.*

This form of wheel is due to Fairbairn. The water is admitted, *below* the axis of the wheel, into curved iron buckets. The wheel is sometimes called a low breast wheel.

3. *Overshot Wheels, in which the Water acts principally by Weight.*

In this class of wheel the water is supplied from the top, and enters the buckets just beyond a vertical diameter. The buckets are formed of a trough shape; they thus hold the water until they have performed nearly one-half a revolution. In the old forms of this wheel the shaft was of cast iron, to which wooden arms were bolted. The water is so regulated that it enters the buckets at a greater velocity than that of the circumference of the wheel, and the water is retained in the buckets as long as possible by means of the form of bucket, and also by a building of masonry called a breast.

4. *High Breast Wheels.*

The high breast wheel is an improvement on the overshot. In this wheel the water is supplied at a point somewhat below a vertical diameter, and is retained in the buckets, partly by their shape, and by a breast which fits the exterior part of the wheel. This wheel in the improved form is constructed entirely of iron, and the water is supplied at a point in the circumference at about 30° from the top of the wheel. The buckets are attached to side plates called *shrouds*, and also to an inner plate called the *sole plate*, the whole being *suspended* from the axis by

means of iron rods, which are secured by gibs and cutters to flanges on the axis. This framework is further braced by diagonal ties or stiffening rods. The sole plates and buckets are usually made of $\frac{1}{8}$-inch plate. It has been found that when the buckets are close a certain amount of water is lost by the compression of the air which takes place on the entry of the water into the bucket, and to obviate this, ventilating buckets, as they are called, have been introduced. In this form of wheel there is virtually no sole plate, the inner edges of the buckets overlapping one another, and secured by bolts in such a manner as to leave a space between them through which the air may find its way. The power is usually transmitted to the machinery by means of a pinion, in inside gear with the wheel, and it has been found that the position of the pitch point should be at or below the centre of gravity of the mass of water in the buckets, i. e. somewhat below a horizontal diameter.

Efficiency of Water Wheels.

The term *efficiency* expresses the ratio between the *useful* work performed and the total energy expended. This ratio is necessarily somewhat varied. In some forms of prime mover the mechanical arrangements are better adapted to utilize energy than in others, and consequently the efficiency is higher in those arrangements.

Class I.—Undershot wheels acting by impulse with flat and radial float-boards, the efficiency is about $\frac{34}{100}$ or $\cdot 34$. In improved forms of this wheel, such as Poncelet's, in which the boards are curved, the efficiency is about $\frac{60}{100}$ or $\cdot 60$.

Class II.—Undershot wheels acting by weight, or low breast wheels. Efficiency $= \frac{65}{100}$ or $\cdot 65$.

Class III.—Overshot wheels. Efficiency $= \frac{60}{100}$ or $\cdot 60$.

Class IV.—High breast wheels. Efficiency $\frac{75}{100}$ or $\cdot 75$.

Speed of Water Wheels.

It has been found that from 4 to 6 feet per second of circumferential velocity is best suited for water wheels, and the velocity of the supply water at about double this.

The total energy of the water acting upon a vertical wheel of the overshot, high breast, or low breast class, is the sum of two quantities. 1st, the energy of the water supplied to the wheel; and 2nd, the energy of the same water due to its position, i.e. in passing from a higher to a lower part of the wheel.

Let v = velocity in feet per second of supply water, and let W = total weight of water supplied per second, and let h = height through which this water passes by the motion of the buckets; then the expression for the total energy is $= \frac{W v^2}{2g} + W h$ or $W\left(\frac{v^2}{2g} + h\right)$, and if we allow for losses by friction, &c., we have as the available energy for performing useful work, $W\left(\frac{v^2}{2g} + h\right) \times$ efficiency; and the effective horse-power of the wheel will be

$$\frac{W\left(\frac{v^2}{2g} + h\right) \times \text{efficiency}}{550}.$$

Example.—A breast wheel of 20 feet diameter is supplied with a weight of water per second $= 2995 \cdot 2$ lbs., the velocity of supply is 12 feet per second. The water is allowed to enter the buckets at a point in the circumference situated 30° from the top of the wheel; the vertical height, therefore, through which the water will act by gravity $= 18 \cdot 66$ feet. The efficiency of the wheel is supposed to be $\cdot 7$. The previous formula thus becomes

$$\frac{2995 \cdot 2 \left(\frac{144}{64 \cdot 4} + 18 \cdot 66\right) \cdot 7}{550} = 79 \cdot 63 \text{ horse-power.}$$

II.—*Horizontal Water Wheels, or Turbines.*

Turbines have been divided into three classes:

1st. Those in which the water passes through the wheel in the direction of the axis.

2nd. Those in which the water passes through the wheel horizontally, and *from* the centre outwards.

3rd. Those in which the water passes through the wheel horizontally, and *towards* the centre. The latter are sometimes called vortex wheels, and are due to Professor James Thomson.

The general arrangement of a turbine is such that the water is guided upon a series of curved vanes set in a radial direction, the curvature of the vanes being such that the water upon leaving them is left with very little energy.

An important form of turbine, called from its inventor the Leffel Wheel, has lately been used with great success in America. From the nature of its action it may be called a double turbine.

The water enters laterally and acts horizontally on

curved vanes; it then passes downwards to another set of curved vanes on which it acts vertically. The wheel rests on a pivot, and is supported by a shoulder on the vertical axis, and from being immersed in the water the displacement goes far to counteract the friction of the supports.

As in the vertical water wheel, the water should be applied to turbines with as little shock as possible, and in order to this the water should glide upon the vanes, and thus produce the necessary pressure or impulse required to drive the wheel.

The efficiency of turbines appears to range from ·65 to ·80, or on an average about the same as for a breast wheel. In some situations they are more advantageous than the vertical wheel, as they are not affected by height of *fall*, and are of less bulk than the vertical wheel.

Reaction Wheel.

In this form of turbine the water issues as a jet from arms, which radiate from a central hollow shaft. These arms may be either straight or curved. The most elementary form of this machine is that known as Barker's Mill. The loss by fluid friction in this form of turbine is considerable. The action of this machine is due to the unbalanced pressure opposite the orifice from which the water escapes, and it is evident that this pressure will vary directly as the square of the velocity of the issuing water, because the pressure of fluids is $= \frac{W V^2}{2g}$, where $W =$ weight of a cubic foot of fluid, and $V =$ velocity of jet in feet per second.

WATER-PRESSURE ENGINES.

In this arrangement the pressure of water due to its head is made use of instead of utilizing the weight of the water or its impulse, as in rotatory arrangements such as water wheels or turbines.

The water-pressure engine is therefore designed in a somewhat similar manner to the steam engine, viz. that by means of a cylinder and suitable valves worked by the machinery, the piston and attached rod which is fitted in the cylinder has a reciprocating motion imparted to it from which the desired rotatory motion is obtained. As in the case of the other prime movers which utilize the energy of water, the design must be such as to reduce the loss of energy by sudden shocks and by sudden enlargement in the supply pipes and channels to a minimum.

HYDRAULIC RAM.

In this machine the water first spends its energy in lifting a heavily formed valve, and which by its subsequent fall through the intervention of a relief valve communicating with a chamber containing air, causes sufficient reaction upon the supply water to cause a portion of it to enter this chamber, and by causing compression of the contained air, sufficient pressure is produced to force part of the contained water upwards through a tube which enters the chamber.

For the working of such a machine as the hydraulic ram an abundant supply of water at a low head is sufficient, and by this means a small quantity of water can be raised through a considerable height.

The advantage of this machine may be expressed by the symbols. $P h = r H$ where $P =$ a great *effort* exerted

through a small head, or distance h; and r = a small resistance overcome through a great height H. The efficiency is said to vary from ·3 to ·6.

Ex.—Let P = 5000 lbs. = supply water,
„ h = 9 feet = fall,
„ r = 700 lbs. = water to be raised,
„ ·5 = efficiency.

Then $5000 \times 9 \times \cdot 5 = 700 \times H$, and $\therefore H = 32$ feet.

WINDMILLS.

Machinery is occasionally driven by wind power, through the medium of vanes or sails in a wheel-like form on which the wind acts and imparts a rotatory motion, which by means of shafting is communicated to the machinery. The shaft carrying the vanes is usually set at an angle to the horizon of about 10°, and can be turned so that the wheel faces the wind. The arms of the sails are fitted with a ladder-like arrangement of pieces of wood on which boards or canvas is laid. The framework of the sail is set at varying angles with the plane of revolution, being flatter at the tips than near the axis, the mean angle or *weather* being about 17°. Since the weight of wind acting on the sail will be as the velocity, and as the energy of this weight of wind is as the square of the velocity, then the power of the mill will vary directly as the cube of the velocity of the wind.

From experiments made by Smeaton, it appears that the effective power of a windmill with sails of about 31 feet diameter, and acted upon by the wind at a velocity of 13 feet per second, was about one horse-power; from this we may determine a general expression of the effective horse-power of such mills.

In the example, the diameter being 31 feet, the area of

a disc of this diameter (which represents the face of the moving body of air acting on or passing through the sails) is 754·66 square feet. The velocity of the wind per second being 13 feet, and if the weight of a cubic foot of air be taken at ·075 lb., then the total energy of this volume of wind per second is $(754 \cdot 66 \times 13 \times \cdot 075) \times \frac{13^2}{64 \cdot 4}$, and total horse-power exerted by wind on sails, is

$$\frac{754 \cdot 66 \times \cdot 075) \times 13^3}{550 \times 64 \cdot 4} = 3 \cdot 51;$$

but from the experiment only one horse-power of effective work was obtained, therefore the ratio of the useful to the total work is as 1 : 3·51, and the efficiency therefore

$$= \frac{1}{3 \cdot 51} = \cdot 28.$$

It follows from the above that the effective horse-power of a windmill may be expressed in terms of the diameter of sail and velocity of the wind;

$$\text{or, effective horse-power} = \frac{D^2 v^3}{2{,}111{,}317},$$

where D = diameter of sail in feet, and v = velocity of wind in feet per second.

NAVAL ARCHITECTURE.

EQUILIBRIUM AND STABILITY OF FLOATING BODIES.

In considering this subject we have:—

I. That as no two bodies can occupy the same space at the same time, there must be a displacement of matter in order that one body may take the place of another.

In the case of a solid body wholly immersed in a fluid,

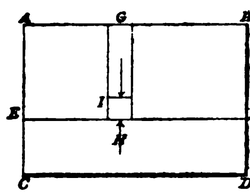
we have a displacement of a mass of fluid of equal bulk to that of the solid body; this fluid is removed or set aside to make room for the solid body.

If the solid body be only partly immersed in the fluid, then the displacement of fluid will be equal to the bulk of the immersed part of the body which displaces it.

II. There will be an *upward* pressure exerted upon the immersed body, and this upward pressure will be equal to the weight of water displaced; and this pressure will act upwards, in the direction of a line drawn vertically through the centre of gravity of the displaced liquid.

This upward force is termed buoyancy, and the centre of gravity of the displaced liquid is called the centre of buoyancy, or centre of displacement.

FIG. 5.



The vertical line drawn through the centre of buoyancy is the direction of the effort of buoyancy.

Let A B C D represent a vessel containing a fluid. Take any horizontal plane through this body of liquid, such as E F.

First, let G H represent a column of liquid whose base, of an area of one square foot, rests upon, or is part of, the plane E F. The weight of this column is equal to its height in feet, multiplied by the weight of a cubic foot of the liquid, $= w\,h$, when w represents the weight of one cubic foot of the fluid, and h represents the height of column in feet. Now, since the column is at rest it must be supported by a force equal to its weight; the upward supporting force is therefore $= w\,h$. It is evident, then, if we suppose the column of liquid to be changed to a column of solid material, of same weight as before, that

we have the same conditions of support; or, as already stated, the solid body will be supported by an upward force equal to the weight of the displaced liquid, or $w h$. Any part of this column will also be in equilibrium, as, for example, the part I H, which may be taken as representing one cubic foot, will be supported by a force equal to its own weight.

If the cubic foot of solid material, I H, be lighter or heavier than the surrounding liquid, then it is evident that it will be impelled upwards or downwards by the difference of pressure due to the weight of liquid and to the weight of the body. Let w, as before, represent the weight of a cubic foot of the liquid, and let w_1 represent the weight of a cubic foot of the solid material; then if w be greater than w_1, there will be a resultant upward pressure exerted on the body of $w - w_1$.

Again, if w be less than w_1, there will be a resultant downward pressure exerted on the body of $w_1 - w$. The difference, therefore, of the specific gravities of the liquid and immersed body will represent the resultant pressure upon the immersed body.

Bodies are said to be in *stable* or *unstable* equilibrium according as they tend to preserve their original position or to depart from it when acted upon by any disturbing force.

These conditions depend on the position of the centre of gravity of the body.

In the case of a wholly immersed homogeneous body, it is evident that the stability of the body will be indifferent, because since the centre of gravity of the body is just the centre of gravity of the displaced fluid, the upward and downward pressures are applied at the same point.

If the body be not homogeneous, then stable equilibrium

can only ensue when the centre of gravity of the body is *below* the centre of gravity of the displaced liquid.

When the body floats so as to present a part above the liquid supporting it, then stable equilibrium may ensue, although the centre of gravity of the floating body be *above* the centre of gravity of the displaced fluid, or centre of buoyancy.

Let A B C D, Fig. 6, represent a rectangular homogeneous solid body, floating so as to have one half of its bulk immersed, the line of flotation, or surface of fluid in which the vessel floats, being E F; then the centre of gravity of the body will be at the centre of figure, and upon the line of flotation, as at O.

FIG. 6.

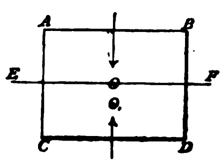
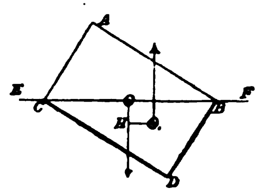


FIG. 7.



The centre of buoyancy, or centre of displacement, will be at the centre of gravity of the immersed part of the figure, as at O_1; and the upward pressure through that point will be equal and opposite to the downward pressure through O, due to the weight of the body.

Let the body now be turned so as to cause it to float, as shown in Fig. 7. The centre of gravity of the body remains as before, but the centre of buoyancy is now shifted outwards, and lies in the centre of gravity of the triangular immersed part, C B D. The body is therefore under the action of a *couple*, tending to restore it to its original position, as shown in Fig. 6. The equal and opposite forces of this couple are the weight of the body, and its equivalent,

the weight of the displaced fluid; and the arm at the ends of which these forces act is the perpendicular distance between the lines of action of these forces, and represented by the line $H O_1$ in Fig. 7.

The body is, therefore, in a state of stable equilibrium, because of the tendency of the body to right itself when, by the action of any force, it is caused to turn round an axis.

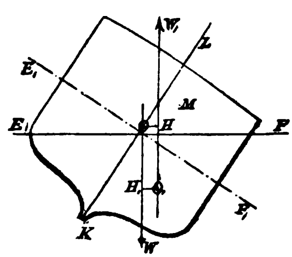
If the body had been of such a form that, when an overturning force was brought to bear upon it, the centre of gravity moved outwards faster than the centre of buoyancy, then it is evident that the result would be the reverse of the first case, because the action of the couple would be to cause still further rotation. The body in this case would, therefore, be in unstable equilibrium.

The relative positions, then, of the centres of gravity and of buoyancy will determine the conditions of equilibrium of the floating body.

Application of these Principles to Ships.

Let Fig. 8 represent a cross section of a ship heeled over at any angle, and let O and O_1, as before, represent the centres of gravity and of buoyancy.

Fig. 8.



The vertical lines W and W_1 are the lines of action of the equal and opposite forces due to weight and displacement. The condition of equilibrium is stable, and the righting couple is $W_1 \times O_1 H_1$.

The point M, where the lines $O_1 W_1$ and L O K intersect, is called the *metacentre*. The line L O K is a line drawn through the centre of

gravity of the vessel or floating body, and the centre of buoyancy when in the original position, or before being heeled over.

The value of the righting couple may also be expressed in terms of the angle of heel; because, by considering Fig. 8, it is evident that, if the line L K be originally perpendicular to the water-line E F, then, when the vessel is heeled over at any angle, say $F\,O\,F_1$, the line L K will now make a similar angle with its original or vertical direction; and, therefore, since the line $O_1\,W_1$ is drawn vertically, the angle $W_1\,M\,L$, or its equal O M H, is equal to the angle $F\,O\,F_1$; and since $M\,O \times \sin. \angle\, O\,M\,H = O\,H =$ arm of righting couple, then $M\,O \times \sin. \angle\, O\,M\,H \times W_1 =$ righting couple.

It is thus evident that, if we multiply the weight of the vessel (or its equivalent the weight of the displaced fluid) by the distance between the metacentre and centre of gravity, and by the sine of the angle of heel, the result will be the moment of stability of the vessel.

The metacentre is sometimes defined to be a point in the "plane of symmetry," which is cut by the vertical line drawn upwards through the centre of buoyancy. The plane of symmetry is the plane passing longitudinally through the line L K; and is thus sometimes termed the "middle-line plane."

The metacentre is also sometimes defined as the centre of curvature of a surface called the "surface of buoyancy," this surface of buoyancy being the locus or surface formed by the centres of buoyancy at different inclinations or angles of heel. When this surface follows any well-known curve, the metacentre may easily be found.

The metacentric height is the distance between the metacentre and the centre of gravity.

The metacentre differs in position.

For rolling, its position generally varies from the water-line to 20 feet above it.

For pitching, from 70 feet to 400 feet above the water-line.

The phenomena in connection with the metacentre have been illustrated by means of pendulums.

In this illustration the point of suspension of the pendulum represents the metacentre, and the motion of the weighted part, or "bob," of the pendulum will represent the rolling of the ship.

Stability or Stiffness.

These terms are sometimes used as synonymous with the term steadiness, but when so used a mistaken idea is apt to arise.

Vessels which have a large amount of stability are found to be deficient in steadiness, and vessels which are noted for steadiness are found to be deficient in stability.

The term stability, as used in naval architecture, means the value or importance of the force resisting change of position, as when the ship is inclined; and the speed with which the ship returns to her original position may be looked upon as the measure of the stability.

Stability may be divided into two kinds, viz. Statical and Dynamical.

1. *Statical Stability.*

The value of this term may be measured by the arm of the moment of stability already explained.

2. *Dynamical Stability.*

This term is used to indicate the work done while the vessel is heeling over, that is to say, the quantity of work

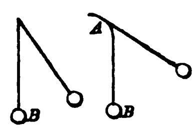
necessary to be performed, in order that the position of the centre of gravity of the vessel may be raised, and the centre of buoyancy depressed.

When, through the action of applied forces, a vessel is made to heel over, the centre of buoyancy must move outwards, or shift its position faster than the centre of gravity, in order that there may be a righting couple. If the centre of gravity move outwards faster than the centre of buoyancy, there will be an upsetting couple.

Stable equilibrium will therefore exist in the first case, and unstable equilibrium in the second case; and for the former, the centre of gravity must be below the metacentre.

If a pendulum be so suspended that when oscillating it winds and unwinds itself on or from a curved surface or "cheek," the "bob" may be made to describe various curves, according to the form of the cheek, and may thus be made to represent the rolling of different ships.

FIG. 9.



If A in Fig. 9 be an arc of a circle, the "bob," B, will describe an involute of a circle, or that curve which is formed by a tracing point fixed to the end of a cord which is being unwound from a circle.

The length, l, of a simple pendulum which will oscillate in equal times with the ship, may be determined as follows: Let g = radius of gyration of the ship, and let d = distance between the metacentre and the centre of gravity (the point of suspension is supposed to be the metacentre), then $l = \frac{g^2}{d}$. This follows from the relations existing in a suspended body between the centres of oscillation, gyration, and gravity. The radius of gyration may also be

expressed as a mean proportional between the length of the simple pendulum and the distance between the metacentre and centre of gravity.

Methods of finding the Metacentre.

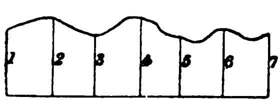
First, to find the centre of gravity of the vessel. The general method adopted is to shift weights through known distances, and to note the angular motion, the displacement being found by calculation.

Professor Rankine's rule, to find the height of metacentre above the centre of gravity is as follows:—

Divide the length of load water-line into equal intervals, at which measure the half breadths at the load water-line; cube each of these half breadths, and regard the cubes as the ordinates of a plane figure having the length of the load water-line as its base. Find the area of that figure by Simpson's rules: Divide two-thirds of that area by the volume of displacement. The quotient is equal to the height of metacentre above centre of buoyancy, from which deduct height of centre of gravity above centre of buoyancy, and the difference = metacentric height.

Simpson's Rule for Areas.

Divide the figure into an *even* number of divisions by a series of ordinates, 1, 2, 3, 4, 5, 6, 7; measure the ordinates, and multiply each by a given multiplier; add the products together, and multiply this sum by one-third of the common interval or distance between each ordinate; the result will be the area of the figure.



The multipliers used for a case as above are 1, 4, 2, 4, 2, 4, 1; the multiplier 4 being always used for the second and second last ordinate of the

series, the multiplier 2 alternating with the multiplier 4 in the intermediate ordinates.

The general expression of the rule is:

$A = \frac{\Delta}{3}(a + 4b + 2c + 4d + \text{\&c.})$, where A = area, Δ = common interval; a, b, c, &c. = value of ordinates.

Rolling.

Easy and Uneasy Ships.—The terms easy and uneasy, as applied to questions in naval architecture, are somewhat indefinite.

The term dipping is used to express the rise and fall of the ship as a whole in relation to the water surface.

When there is a motion of the water, or when the wind comes in gusts, there may result a combination of these movements, and necessarily greatly intensifying their effects on the ship.

It appears that in uneasy ships the axis of angular motion may take up any position, and that vessels having short periods of rolling roll through great angles.

When the centre of gravity is high the ship is steadier than when it is low, vessels carrying cargo low down rolling very much. In the 'Bellerophon' and 'Hercules' the centre of gravity is raised by placing the engines and boilers on raised platforms, the object being to diminish rolling.

In the 'Achilles' the centre of gravity is said to be 3 feet from the metacentre; whilst in the 'Prince Consort' the centre of gravity is said to be 6 feet from the metacentre. The latter vessel is found to roll more than the former.

The great stiffness of the first ironclads caused them to roll considerably.

WAVES.

Waves in section may be considered trochoidal; the exact shape of waves and their particular motions have not however as yet been completely ascertained.

Fig. 11. Fig. 10.

In Fig. 10 the horizontal line A B represents the surface of still water, and if by the action of any force or forces we cause a depression or hollow at any part, such as at H, then a corresponding rise or elevation must follow, as at A_1. The action of this downward force at H may be conceived to be communicated from H to A_1 by producing pressure on the adjoining parts, which becomes transmitted laterally as shown at C, Fig. 11, and in consequence of the uprise of fluid at A_1 the particles of water originally about H will tend to move in the direction D.

During the fall of the particles about A_1, whose motion is represented by D, a combined motion will again result as at F, and finally, after the particles have passed the horizontal line or line of still water, a return motion from left to right as at G will follow.

The particles of water at and near the surface, as at H, may be considered, during the formation of crest and hollow, as moving in curved paths which are considered to be circular or elliptical.

It is evident from the foregoing consideration that wave motion is propagated by a transmission of pressure. The first hollow, say, being formed, a corresponding elevation is produced through the communicated pressure. On the

fall of this upraised fluid a hollow is now formed in advance of the first hollow, and another elevation is produced in advance of the first elevation, and this series of operations will continue until the energy expended in producing the first hollow has been dissipated through friction or resistance to motion of the particles of the liquid.

What is called the "period" of a wave is the time occupied by a particle in making one revolution, and this is evidently the same time as that in which the wave travels its own length, the length of a wave being measured from crest to crest or from hollow to hollow.

The speed of waves varies with their lengths and with the depths of water, and has been stated as follows:—The speed of a deep water wave is about equal to the "velocity acquired by a body in falling through a space equal to half of the radius of a circle whose circumference is the length of the wave;" or, $V = \sqrt{\frac{Lg}{2\pi}}$, where V = velocity in feet per sec. and L = length of wave in feet, and the period of a wave will be $P \overset{\text{secs.}}{=} \sqrt{\frac{L\,2\pi}{g}}$.

Example.—Let L = 600 feet, then

$$V = \sqrt{\frac{600 \times 32 \cdot 2}{2 \times 3 \cdot 1416}} = 55 \text{ feet per sec.}$$

$$\text{and } P = \sqrt{\frac{600 \times 2 \times 3 \cdot 1416}{32 \cdot 2}} = 10 \cdot 8 \text{ secs.}$$

The "breaking" of waves indicates a too rapid motion of the upper particles; this may be due to the action of the wind impelling the particles forward, or to the shoaling of the water where, owing to the decreased depth of water, the pressure due to the fall from crest to hollow

will be transmitted more intensely in a horizontal direction and cause a more rapid movement of the particles whilst rotating and induce distortion of the form of wave surface.

Ocean waves are reckoned to reach to the height of about 40 feet; their lengths being about 600 feet, and sometimes probably 1000 feet.

The pressure of ocean waves when striking a resisting surface, as obtained from experiment, reaches 6083 lbs. per square foot.

Approximate rule, by Stevenson, for height of ocean waves:

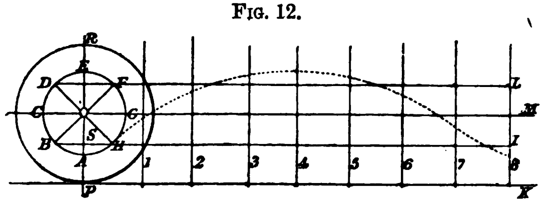
$$h = 1\cdot 5\ \sqrt{\text{fetch in nautical miles.}}$$

The term "fetch" denotes the distance from the point of origin of wave to the point of observation. The mean value of a nautical mile is 1·1508 statute mile.

To Draw a Trochoid.

A trochoid curve is the curve described by a point in a rolling circle situated within the circumference (the curve described by a point in the circumference being a cycloid).

FIG. 12.



The curve may therefore be drawn by fixing a tracing point inside the circumference of the circle. In Fig. 12, R is the rolling circle moving on the line P K. P K is the distance moved over in a revolution. A is a point inside

the circumference of R, which as the circle moves forward takes up successively the positions B, C, D, E, &c.

Draw the lines D, F, L and B, H, I parallel to P K. Let the arcs A B, B C, C D, &c., be $\frac{1}{8}$th of the circle A C E G. Divide P K into eight equal divisions, then when the centre O has advanced $\frac{1}{8}$th of P K, $\frac{1}{8}$th of a revolution will have been made by the point A. So that when O is on the vertical line 1, A will have taken the position B, so that if we set off on the line B I a length measured from the line 1 and to the left of it, and corresponding to B S, this will give us the position of A on B I. Again, when O has reached 2, A will now have reached the position represented by C, and by setting of a length O C measured back from 2, the position of A on C G M will be found. When O has reached 4 one-half revolution will be completed, so that A is now at the position formerly occupied by E, and its position on the line E O will be directly above 4; these are points on the trochoid, so that by joining them the curve may be shown.

*Action of Waves on a Ship.**

Passive heaving is a term applied to conditions such, that the floating body would exactly move in the same orbit as the particles of water. This action combines vertical and transverse movements, as rising and falling and swaying sideways.

When a ship is in smooth water, we may look upon the terms stiffness and steadiness as similar; but when the vessel rolls on waves, stiffness means the tendency to keep upright to the wave surface. Steadiness means the tendency to sit "fair" or upright to a horizontal plane.

* These actions are well described in 'Report, British Association, on Sea-going Qualities of Ships,' 1869.

A stiff ship is generally a "dry" ship, and carries sail well; a steady ship, on the other hand, is a strong ship, and suits well for a war vessel, as the guns can be worked better than in the other case.

Passive pitching or scending are terms used to express the oscillations made by a ship when she moves in an oblique direction to the wave crests.

The oscillations of a ship are due to a combination of motions, viz. those made in time with the waves, and those which would be produced in still water.

When a ship sways or oscillates round an upright axis she is said to "yaw."

A light stiff ship tends to float as a raft does.

The term "between wind and water" refers to those parts subject to immersion and emersion.

Steadiness may be partly secured by a deep draught of water; the reverse is true for stiffness.

Lurching is due to gusts of wind.

The motions of pitching, scending, and yawing may be partly remedied by keeping the ends of the vessel free from heavy weights.

During rolling, the most dangerous movement is towards the crest of a wave.

When the waves come "broadside on," the ship is in the worst position for receiving their impact. With the waves "head on," the apparent wave-period is shortened; when "head off," it is lengthened.

Instruments for Measuring Roll.

These instruments are the common pendulum, the clinometer, or slope measurer, and an adaptation of the gyroscope.

Both the pendulum and clinometer are indifferent

instruments for roll measurers, as they tend to prolong their angular deviation, so that the rolling motion as indicated is too great.

The gyroscopic instrument consists of a rapidly rotating body having its centre of gravity situated below the point of support. An agate cup supports the spinning body, and the top of the spindle carries a pencil which marks a band of paper driven by clockwork.

Stream Lines.

These are lines traced by a particle of a steady current of fluid. Professor Rankine, who mathematically investigated these lines, calls them neoïds or "ship-shape" lines. "Stream line surfaces" are those surfaces of a vessel over which the particles of fluid glide easily; such surfaces are termed "fair." The fluid does not form eddies, but glides along smoothly.

Resistance to the Motion of a Vessel.

It has been established by Professor Rankine that for wholly submerged bodies with fair lines, and also for blunt-ended ovals, the surface friction is the principal resistance.

The various resistances offered to a ship on her passage through a fluid, such as water, have been classed as follows:—

1. Head Resistance. 2. Skin Resistance. 3. Back Pressure.

1. *Head Resistance.*—By the forward motion of the vessel waves are produced, and in order to set up these oscillations of the water, energy must be spent.

2. *Skin Resistance.*—When a vessel moves forward, a

certain quantity of water is carried along by the friction of the skin or outward surface of the submerged part of the ship.

3. *Back Pressure.*—This is due to the water which follows the ship filling up the space which she occupied; it is evident that this water flows in from behind, beneath, and from the sides. If the form of the vessel be not what is called "fair," eddies and other waves are produced; when a vessel moves forward, the first resistance experienced is that due to "head pressure," which forms what is termed a "head swell." If the vessel be driven quicker than the motion of the swell, she will tend to rise upon it; generally speaking, the total resistance varies as the square of the speed.

In Mr. Scott Russell's form of least resistance, we have the bow a curve of sines (or harmonic curve), the middle straight, and the after part trochoidal. Professor Rankine treats this question of resistance as follows:—

1. The distortion of the particles of water.
2. The production of currents.
3. The production of waves.
4. The production of frictional eddies.

The first is assumed to be without effect on ships, and need only be taken into account in experimenting with models.

The second does not affect a ship whose surface is fair, because the particles of water glide over that surface.

The third shows that there is for each vessel a limit of speed, above which the resistance due to this cause is greatly increased.

The fourth is due to the adhesion between the water and the ship's surface, and also to the viscosity of the water. It has been found that this frictional resistance is

proportional to what is called the "augmented surface," and not to the actual immersed surface.

The augmented surface has been expressed as follows:—

Let s = actual surface and q = velocity ratio of gliding,* then $\int q^3\, d\, s$ = augmented surface.

The eddy resistance also has been expressed as follows:—

Let V = speed, g = gravity, and w = weight of a cubic foot of water, and f = coefficient of friction, then

$$V^2 f \frac{w}{2g} \int q_3\, d\, s = \text{eddy resistance.}$$

For clean painted surfaces f is taken as equal ·004.

It appears that at 10 knots speed the eddy resistance is = 1 lb. per square foot of augmented surface.

The term $\int q^3\, d\, s$ is supposed to include the skin resistance, and the resistances of the elevation and depression of the water at bow and stern.

It has also been determined, that "the augmented surface and its resistance has been found to be the same as that of a trochoidal riband, whose length is the length of the boat on the plane of flotation, and whose breadth is the mean immersed girth of the vessel." To find the augmented surface of a trochoidal riband of a given base and of a given breadth, we multiply the base × breadth × coefficient of augmentation.

This coefficient of augmentation is equal to

$$1 + 4 \sin.^2 a + \sin.^4 a,$$

where a is the angle of greatest obliquity to the horizon made by a tangent to the trochoid.

* The term velocity ratio means the ratio which two velocities bear to one another.

Professor Rankine's coefficient of propulsion is:—"The augmented surface in square feet, multiplied by the cube of the speed in knots, and divided by the indicated horse-power."

For clean iron vessels this coefficient = 20,000.

For copper-sheathed " " = 21,800.

The above expressed as a formula will be

$$\frac{\text{Length of plane of flotation} \times \text{mean immersed girth} \times \text{coefficient of augmentation (or } 1 + 4\sin.^2 a + \sin.^4 a) \times V^3 \text{ in knots}}{\text{Indicated horse-power}}.$$

From experiments by Mr. Froude on towing resistance of the full-sized ship 'Greyhound' (coppered vessel), 1157 tons, it appears that

at 4 knots the resistance of this ship was approximately			=	0·6 ton.
6	"	"	"	1·4 "
8	"	"	"	2·5 tons.
10	"	"	"	4·7 "
12	"	"	"	9·0 "

PROPULSION OF VESSELS.

The effect of a propeller is to produce an action on the water, and hence by the reaction of the water on the propeller the vessel is propelled forward.

The simplest form of a propeller is that of oars. In this case, the oars being on each side of the boat, a current of water is driven backwards from each oar-blade; the reaction of the water of this current takes effect on the boat through the thrust at the rowlocks.

We have, secondly, paddles or side-wheels. These act similarly to oars in driving back lateral currents of water. In the screw propeller, as applied at the stern of a vessel, a single current of water is sent backwards.

In the hydraulic propeller a jet or stream of water is caused to issue from the sides of the vessel; the impulse of

this moving water acting on the still water without produces a forward movement of the vessel.

The term slip is used to denote the difference of the velocity of the propeller and the velocity of the ship.

Positive slip is that slip which takes place when the velocity of the propeller is greater than the velocity of the ship.

Negative slip is that slip which takes place when the velocity of the ship is greater than the velocity of the propeller.

Positive slip is that which generally takes place during the action of a propeller.

Negative slip is a phenomenon occurring frequently with screw propellers, and has been variously accounted for.

It has been thought that since the screw propeller is so situated that it takes effect on the water of the wake, or current of water which follows the vessel, the effect of the propeller would be increased.

Another explanation is that the motion of the particular following wave crest (that is, the wave which fills in at the stern) is affected by the screw-blade. This arrest of velocity in the wave causes loss of power.

Reaction of Propellers.

The pressure on the propeller or float-board is due to the reaction of the impulse exerted by the board on the fluid.

We have four cases which may be considered:

I. When the water moves faster than the board.

II. When the water moves at the same rate as the board.

III. When the board moves faster than the water.

IV. When the water is at rest and the board is in motion.

I. Let v_1 represent the velocity of the board, or of velocity of water after contact, and v_2 the original velocity of the water, both in feet per second; and let w = weight of a cubic foot of water, and n the quantity of cubic feet of water acting per second. The relative velocity of board and water will be $v_1 - v_2$, and $n\,w\left(\frac{v_1 - v_2}{g}\right)$ = impulse or pressure due to n cubic feet of water, i.e. the impulse is equal to the change of momentum, for

$$n\,w\frac{v_1}{g} - n\,w\frac{v_2}{g} = n\,w\left(\frac{v_1 - v_2}{g}\right).$$

II. In this case $v_1 = v_2$, therefore $n\,w\left(\frac{v_1 - v_2}{g}\right) = 0$; that is to say, there is no impulse exerted. This is evident when we consider that the board is virtually floating in the water; also, since there is no change of momentum, there can be no impulse exerted.

III. In this case we have $v_1 - v_2$, of which v_1 is the greater, and the expression for pressure or impulse is $n\,w\left(\frac{v_1 - v_2}{g}\right)$, which is the same as that in case No. I., but in this instance the result is positive, indicating that the impulse is exerted in an opposite direction. In case I. the impulse is exerted from the water to the board; in case III. the impulse is exerted from the board to the water.

IV. In this case, $v_2 = 0$, and therefore we have $n\,w\frac{v_1}{g}$ = impulse exerted = momentum produced.

Case No. III. is applicable to the propeller of a vessel. v_2 in this case is the speed of the vessel; $v_1 - v_2$ is

therefore equivalent to the slip of the float-board of the propeller.

Now, to form a rule whereby we may calculate the thrust of a propeller, we have

v_1 = velocity of float-board in feet per second.
v_2 = velocity of vessel in feet per second.
w = weight of one cubic foot of water.
n = number of cubic feet of water acting per second.

In this example n is evidently equal to v_1, because the board is in contact with the water.

$w\,v_1$ is therefore equal to quantity of water; the impulse then will be $w\,v_1\left(\frac{v_1 - v_2}{g}\right)$, and therefore we have the following rule for the thrust of a propeller in lbs. = T lbs. = 2 × area in sq. ft. × (velocity in feet per second of stream from propeller) × slip in feet.

Example.—Let $v_1 = 20 \cdot 5$ feet per second = velocity of periphery of wheel, or centre of pressure of float-board; $w = 62 \cdot 4$ lbs. for fresh water, and 64 lbs. for sea-water; $g = 32 \cdot 2$; $v_2 = 15 \cdot 5$ feet per second; area of float-boards = 40 square feet. Then if 64 lbs. be taken for value of w, we have $\frac{64 \times 40 \times 20 \cdot 5 \times 5}{32 \cdot 2} = 8149$ lbs.

Professor Rankine gives the following rule for the thrust of a propeller in lbs. for paddle, screw, or jet:—

"Multiply together the transverse sectional area in square feet of the stream driven astern by the propeller; the speed of that stream *relatively to the ship*, in knots; the *real slip*, or part of that speed which is impressed on that stream by the propeller, also in knots; and the constant 5·66 for sea-water, and 5·5 for fresh water."

From experiments by Col. Beaufoy, it appears that at a

speed of 12 knots per hour 1 square foot had a resistance of 350 lbs.

Experiment with H.M.S. 'Rattler,' with a screw of 10 feet diameter, 11 feet pitch, 104 revolutions per minute. Speed due to propeller, $11\frac{1}{2}$ knots; area of screw's disc = $78 \square'$ and 78×350 lbs. = 12 tons as a pressure to shaft. Actual thrust by dynamometer = 4 tons 3 cwt., from which Mr. Griffiths thinks that the quantity of water supplied to the screw had been only half of what was required.*

Paddle-wheel Propellers.

The float-boards of paddle-wheels are either radial or feathering. When the floats are fastened on radially, a great loss of power takes place on account of the obliquity on entering and leaving the water.

By means of the feathering wheel the float-boards enter, continue in, and leave the water nearly at right angles to its surface; this is effected by means of a series of rods, placed eccentrically, or radiating from a centre which is not the centre of the wheel. These rods at their outer extremities are attached to short arms fastened to the float-boards; the float-boards and their arms being movable round pivots in spurs projecting from the rim of the wheel, have their positions changed as the wheel rotates, the adjustment being made so as to enable the float to act upon the water, as stated above. Float-boards are immersed more or less deeply, according to the kind of vessel.

The centre of pressure of feathering float-boards may be considered as situated at the middle of the boards; in radial floats it is farther out, and may be taken as situated at about one-third of depth of float from outside edge.

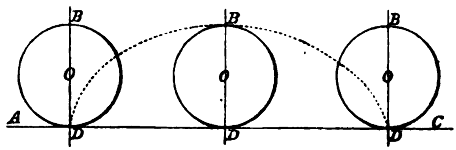
In connection with paddle-wheel propellers the term

* Lecture by Mr. R. Griffiths, Royal United Service Inst.

rolling circle is sometimes used. This rolling circle is the circle described by a point in the wheel which would generate a cycloid, or, in other words, the circle described by a point in the wheel whose *rotatory* motion is equal to the speed of the vessel.

In Fig. 13, the circle B D, in rolling along the plane

FIG. 13.



A C, generates the cycloidal curve D B D, D D is therefore equal to B D $\times$ 3·1416, and this is the horizontal distance moved through by the centre of rolling circle O. We may thus form a rule for determining the diameter of the rolling circle when the speed of the vessel is known. Let d = diameter of rolling circle, v the velocity of the ship, represented by D D, and n the number of revolutions; then $d = \frac{v}{n \times 3 \cdot 1416}$.

If v = number of knots per hour, and n = number of revolutions per minute, then the diameter in feet =

$$d = \frac{v \times 6080}{n \times 3 \cdot 1416 \times 60} = \frac{v \times 32 \cdot 25}{n}.$$

The difference between the rotatory velocity of a point in the rolling circle and of the rotatory velocity of the centre of pressure is evidently the effective velocity in overcoming the resistance of the vessel, and is simply the slip of the propeller.

Screw Propellers.

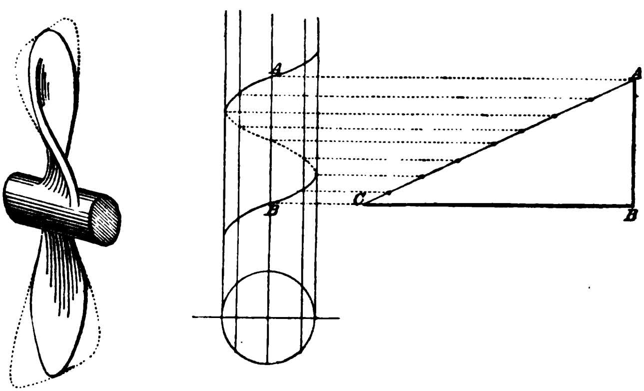
Screw propellers have from two to four blades. Those propellers having two blades appear to give good results so far as speed is concerned, but propellers with three and four blades give a much steadier motion and less vibration.

The pitch of propellers is of different degrees of fineness or coarseness.

The blades of screw propellers are curved to such surfaces that, if continued, they would form a helix or screw, the central line or axis being the axis of the shaft on which the propeller is fitted.

In Fig. 14, the curved lines round the cylinder represent a screw or helix, the distance A B being the pitch or

FIG. 14.



the distance between two similar points in the *same* thread. Now, if we suppose this part of the thread to be unwound, we shall have the inclined plane C A, in which C A = length of thread from A to B, that is of one turn, and A B is the pitch, so that if d = diameter of cylinder on which the screw is cut, then the pitch $= p = d\,\pi$ tan. a,

where a = angle which the thread makes with a perpendicular to the axis, and $\pi = 3 \cdot 1416$.

The pitch × number of revolutions is the speed due to propeller.

In the Griffiths' form of propeller a large boss, about one-third of the diameter, is introduced at the centre, and the blades are tapered to their extremities.

About one-half only of the power of the engine is utilized by either screw or paddle propeller. Different views have been expressed as to the accounting for this great loss of energy, even after allowing for friction of the machinery. According to some, in screw propulsion there is an insufficiency of water for the screw to act upon. Others believe that it is accounted for by the reaction of the propeller on the stream lines at the stern.*

According to some, the resistance which a screw propeller meets with increases with the depth of water. Some, however, are of opinion that any variation of resistance of the water is due to the presence of air, this air being drawn in by the screw. The presence of the air is apparent by the foam which is raised by the propeller.

Hydraulic Propellers.

Ruthven's propeller acts on the principle of the turbine, the water being drawn into the wheel or turbine, and ejected forcibly through curved openings in the sides of the vessel.

Horizontal Propellers.

In Moodie's propeller, tried in some small steamers on the Clyde, a vertical shaft carries a T-shaped bracket, at or near the extremities of which work two floats, each turning on a vertical axis. The whole is put in rotation

* See Report Exp. Resistance of a Full-sized Ship, Froude, 1874.

by the engine. The float motion is produced by spur-gear, and the conditions are such, that in one whole revolution of the central shaft and bracket, one half revolution of the floats takes place. The result of such motion is, that when one float is in a position at right angles to the line of the keel, and therefore most favourable for propulsion, the other float is in the same line as the keel, and is therefore offering little or no resistance to the rotation. The vessel is therefore pushed forward.

Experiments made showed a gain in favour of this propeller over the screw propeller of about 35 per cent.*

The greater number of moving parts, however, is somewhat prejudicial to its adoption for sea-going vessels.

CONSTRUCTION OF VESSELS.

A ship may be considered to a certain extent as a beam or girder. Girders, however, are designed to bear loads acting always in a given direction. The case of a ship amidst waves presents a very complex problem. At one time the bow and stern may be the principal points of support, at another the middle parts of the vessel may be the main support. Again, in rolling, the sides will be brought alternately into positions in which they will be subjected to different conditions of strain.

Probably the best comparison that can be made is that of a hollow or tubular girder, such as the Britannia Bridge, or the Victoria Bridge at Montreal, where the top and bottom flanges will represent respectively the deck and bottom of the ship, and the side-plating or webs of the girder will represent the sides of the ship.

The longitudinal strength should be continuous, and not broken, as is sometimes the case at the bulkheads.

The rigidity of the whole structure should be ensured.

* See 'Trans. Inst. Engineers and Shipbuilders in Scotland,' vol. xii.

The deck beams give transverse strength, and enable the vessel to resist the strains due to the motion in a sea-way.

The principal parts to be considered are:

1st. The keel.

2nd. The stem and stern-post.

3rd. The frames, transverse and longitudinal.

4th. The deck.

5th. Pillars.

6th. Plating and riveting.

1st. The keel. Keels were at one time placed externally. They have been designed of various forms. Internal keels are now of frequent use.

2nd. Stem and stern-post are generally made of iron bars, and welded to the keel.

3rd. The frames, transverse and longitudinal. In transverse framing we have a series of reversed angle-iron bars set athwart the vessel, as ribs to which the skin plating is attached.

Longitudinal, the vessel is divided by bulkheads, with longitudinal stringers between, these longitudinal stringers being made somewhat like the "transverse frames."

4th. Decks. The deck is of great importance. Deck beams stretch across and tie the frames together; on these beams wooden planking is laid. A sufficient amount of plating should be introduced between the planking and the deck beams to make the deck act as the upper flange of a girder. These beams are affected by both compressive and tensile strains.

5th. Pillars are placed above the keel; they support the deck beams, and, if properly connected, also act as ties.

6th. Plating and riveting. The plating is usually placed so that one plate is overlapped by the plates above and below it. The rivets are countersunk, so that their heads are flush with the surface of the plating.

There are two principal forms of riveting known as zig-zag and as chain riveting. In the former the rivets are arranged triangularly, whilst in the latter they are arranged in parallel rows.

Composite ships are those in which wooden planking is fastened to iron frames.

In the design of vessels it is necessary to distribute the load in reference to the displacement, so that there may be little or no tendency towards the action of unbalanced pressures, the tendency being towards a deficiency of buoyancy at the stem and stern, and to have an excess of buoyancy amidships. This excess of buoyancy amidships, or upward pressure due to the displaced liquid, tends to produce a curvature in the water-line of the vessel, which is convex upwards. Too much weight placed amidships produces a curvature in the reverse direction. The first-mentioned change of figure is technically termed *hogging*, and the latter *sagging*.

The bending moment arising from these actions will vary with the form of vessel, on an average it may be $= \dfrac{\text{displacement} \times \text{length}}{20 \text{ to } 30}$. The moment of resistance of the cross section (taken at amidships) must be equal to this bending moment. The moment of resistance is found by finding the moment of inertia of the ironwork in the cross section, and multiplying the latter by the ratio of safe strength of iron, to distance of most severely strained plate of iron from neutral axis (or centre of gravity of section). Let M = bending moment, and let I = moment of inertia of the cross section, let f = greatest safe stress on the iron, and let y = distance of that piece of iron from neutral axis, then $M = I \times \dfrac{f}{y}$.

The distance y or depth of neutral axis from deck plating will probably be about $\frac{4}{7}$ths of the total depth or distance from deck to bottom plating.

Forms of Vessels.—The old practice was to make the length equal to four times the beam, and very full forward, or with what was technically called a "cod's head and mackerel's tail."

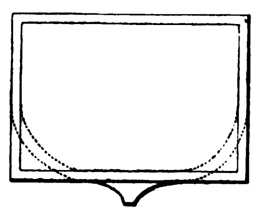
Iron screw steamers are now made of lengths from ten to fourteen times the beam, the depth about $\frac{1}{12}$th the length, and drawing a considerable deal of water to suit the propeller.

The weights of a large ordinary merchant steamship having a displacement of 5000 tons bear about the following proportions:—

	Per Cent. of Displacement.
Vessel	32
Engines and boilers	5
Coals	9
Cargo, &c.	54
	100

To give an example of the manner in which we may generally show how the strains in a ship may be calculated. Let the vessel be considered as a hollow beam, rectangular in section, the shell of which for simplicity is supposed to be of a uniform section. Let the vessel be 320 feet in length, 34 feet broad, and 30 feet deep. Displacement = 3400 tons.

FIG. 15.



Let the greatest bending or hogging moment be taken as $= \frac{\text{displacement} \times \text{length}}{20} = M$. This formula for the

bending moment is identical with that used for a uniformly loaded beam, or $\frac{W \times L}{8}$ because the length in this case is not the extreme length of the vessel, but the assumed distance at which it may be considered that she is supported by two waves passing in line of length: this length in the above case is $= \frac{L}{2 \cdot 5}$ and $\therefore \frac{W \times \frac{L}{2 \cdot 5}}{8} = \frac{W \times L}{20}$. W in this case = displacement, or its equivalent, the weight of the ship.

The above moment must be equal to the greatest moment of resistance, or, as stated at p. 46, $M = \frac{I \times f}{y}$, from which when any three terms are known the fourth may be found.

Let it be required to determine the quantity f, and let an assumed uniform thickness of 1″ of metal be taken for the section.

Consider this case as in example given of calculation of moment of resistance of rectangular hollow beam ("Strength of Structures").

$$A_1 = 34 \times 12 \times 1, \quad A_2 = 34 \times 12 \times 1$$
$$h_1 = 15 \times 12 \qquad h_2 = 15 \times 12$$

(the exact value for h_1 and h_2 is evidently $(15 \times 12) - \cdot 5$; but in the present example this difference will not materially affect the result).

$$b = 1, \text{ and } d = (30 \times 12) - 2$$
$$I = (34 \times 12 \times 1) \times (15 \times 12)^2 \times 2 + \frac{1 \times 358^3}{12} \times 2$$
$$= 26438400 + 7647118 = 34085518$$

$$\text{Moment of resistance} = \frac{\mathrm{I}f}{y} = \frac{34085518 \times f}{15 \times 12} = 189363f.$$

Now to find the value of f, we have

$$\frac{\text{Displacement} \times \mathrm{L}}{20} = \frac{\mathrm{I}f}{y} = \frac{3400 \times 320 \times 12}{20 \times 189363} = f$$
$$= \frac{652800}{189363} = f = 3 \cdot 44 \text{ tons per square inch.}$$

3·44 tons per square inch is therefore the greatest stress to which the metal of such a section would be subjected.

Seeing that wrought iron resists tensile stress better than compressive stress, the section may be adapted to suit this variation, but it must be borne in mind that the part of the section in tension is not wholly effective, on account of loss of material through fastenings, such as rivet-holes. It may happen, therefore, that the effective section in tension is less than the actual section, in the same proportion as the tensile is greater than the compressive strength of the material; thus a uniform coefficient of strength may be taken in computing the moment of resistance.

If we have reason to suppose that the vessel is liable to bending action, in such manner that the sides now act the part of upper and lower members, whilst the latter mentioned parts have now taken the position of the sides, as might approximately happen in heavy rolling, then we must make a second calculation for the new moment of resistance round its axis perpendicular to the one taken in the former case.

The actual number of square inches in the section of a vessel of the dimensions given shows that the mean

equivalent rectangular section would be about ·9 inch thick, and the distribution as follows:—

	Sq. In.
Keel, keelson, &c.	250
Bottom plating..	240
Side plating, &c.	300
Decks, stringers, &c.	530
	1320

The lower parts of the vessel then have	490
Sides	300
Upper parts	530

Where wood is used, its section may be reduced to its equivalent in iron by dividing by 16.

Freeboard is a technical term used to denote the height of the side of the ship above the water-line.

Various rules have been given for calculating this quantity. Mr. Robert Duncan, Past President Inst. Engineers and Shipbuilders in Scotland, in his rule for freeboard, uses the *sum* of the length, breadth, and depth. The *length* is taken between perpendiculars, the *breadth* is the extreme breadth, and the *depth* is depth of hold measured from the upper deck.*

For first-class sailing vessels, the freeboard in inches is $F = (L + B + D) \cdot 2$.

For first-class steam vessels, $F = (L + B + D) \cdot 15$.

Example.—Length 380′, breadth 38′, depth of hold 28·5′, and $380 + 38 + 28 \cdot 5 = 446 \cdot 5$ and $446 \cdot 5 \times \cdot 2 = 89 \cdot 3''$.

* See 'Trans. Inst. Engineers and Shipbuilders in Scotland,' vol. xvii.

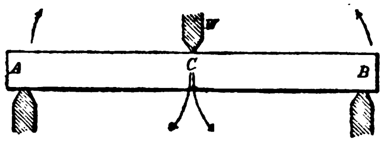
STRENGTH OF STRUCTURES.

STRAINS IN GIRDERS.

Solid rectangular beams.

If the beam A B be supported at the two extremities, and a load W be applied at the middle, there will be a bending action set up in the material of which the beam

FIG. 16.

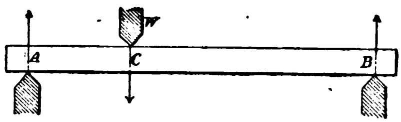


is composed, and there will also be a tendency to cut through or *shear* due to the action of the load and the consequent reaction of the supports.

It is evident that the upward supporting forces must be together equal to, and opposed to the load; and if the load be acting at the centre of span, then each supporting force will be equal to half of the load, or $\frac{W}{2}$; and this will represent the shearing action which must be resisted by the material of the beam. From this it follows that the beam is also under the action of two couples acting in the direction shown by the arrows. The forces of these couples are each equal to $\frac{W}{2}$, and their arms are each equal to $\frac{S}{2}$, where S = span of beam, or distance between supports. The moments of these couples are each equal to $\frac{W}{2} \times \frac{S}{2} = \frac{W \times S}{4}$, and this is called the Bending Moment of the beam.

If the load be not at the centre of the span, then the supporting forces will no longer be equal, but will be inversely as the parts into which the span is now divided. Thus in Fig. 17 the load W is placed nearer to A than

FIG. 17.



to B. The supporting forces are now found by the following proportion:—

$$W : F_A :: S : CB \therefore F_A = \frac{W \times CB}{S} = \text{supporting force at A,}$$

$$\text{and } W : F_B :: S : CA \therefore F_B = \frac{W \times CA}{S} = \text{supporting force at B,}$$

and these values F_A and F_B will represent the shearing forces acting in each division of the beam.

The bending moment to which the beam is subjected will be

$$F_A \times CA = F_B \times CB = \frac{W \times CB}{S} \times CA = \frac{W \times CA}{S} \times CB.$$

If the load be uniformly distributed, so that we have a certain load per unit of span, such as so many lbs. per foot, or $w \times S$, then it is evident that the supporting forces must be each equal to $\frac{w \times S}{2}$; and the greatest bending moment, viz. at centre of span, will be

$$\frac{w \times S}{2} \times \frac{S}{4} = \frac{w S^2}{8} = \frac{W S}{8},$$

where $W = w S$ and $\frac{S}{4}$ is one-fourth of the span, or one-

half of the half span, which is the point at which the resultant of the half load may be assumed to pass.

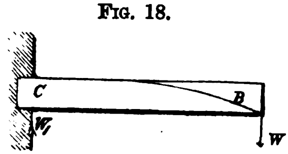
We may find the bending moment by integration as follows: Let w = load per unit of span, and let dx = unit of span, then $w\,dx$ = load on dx. Let x be any distance from centre of span, then $w\,dx \times x$ = moment of that part of load represented by $w\,dx$, so that $w\int x\,dx$ = total bending moment $= \frac{w x^2}{2}$, which, when $x = \frac{S}{2}$, or half span, gives $\frac{w S^2}{8}$ or $\frac{W S}{8}$ as before; and since the greatest bending moment when single load W is applied is $= \frac{W S}{4}$, it follows that for same span and same load the beam is twice as strong when under a uniform load as when under a single load.

When the beam is supported at one end only.

Let Fig. 18 represent a beam supported at W_1, and loaded at its extremity B.

Let W = load at B. The shearing force in C B will be = W, and the bending moment = W × C B = W × L, where L = length of beam. To compare the strength of such a beam with one supported at both ends, we have to consider that, since this beam is in the condition of one half of that in Fig. 16, inverted, then

FIG. 18.



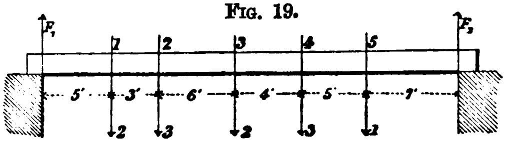
$W \times CB = \frac{W}{2} \times CB$ in Fig. 16; so that if W = same load in both cases, and C B = same length, then the

bending action is twice as great in the latter example (Fig. 18) as in the former (Fig. 16).

If the load be uniformly distributed at w per unit of length, then $w \times \text{L} = $ total load, and the resultant action will pass at $\frac{\text{L}}{2}$ from point of support; the bending moment will therefore be $= w \times \text{L} \times \frac{\text{L}}{2} = \frac{w\,\text{L}^2}{2} = \frac{\text{W}\,\text{L}}{2}$.

When a beam is under the action of a series of loads placed at various distances along its length, the conditions are more complex.

FIG. 19.



Let a load of 11 tons be arranged upon a beam of a span of 30 feet, in such a manner that the loads are as follows:—

	Loads.	Distance from F_1.
At 1	2	5′
2	3	8′
3	2	14′
4	3	18′
5	1	23′
	11	

(1.) To find the supporting forces F_1 and F_2 we must take the sum of the moments of all the forces acting, and divide by the sum of the weights, as follows:—

	Load.		Arm.		Ft. Tons.
M_0 round F_1	= 2	×	5	=	10
M_1 „	= 3	×	8	=	24
M_2 „	= 2	×	14	=	28
M_3 „	= 3	×	18	=	54
M_4 „	= 1	×	23	=	23
	11				139

$$\frac{139}{11} = \frac{\text{sum of moments}}{\text{sum of weights}} = 12 \cdot 63.$$

12·63 then distance from F_1 at which resultant passes, and $\therefore$ 17·37 = other portion of beam. The supporting forces are therefore $\frac{12 \cdot 63 \times 11}{30} = 4 \cdot 631$, $= F_2$, and $\frac{17 \cdot 37 \times 11}{30}$ $= 6 \cdot 369$, $= F_1$.

(2.) To find the bending moments round the points 1, 2, 3, 4, and 5.

The forces acting at the various parts of the beam are:

	Upwards.	Downwards.		Resultant.	Moments.	Sum of Moments.	Total Moments at
At F_1	6·369	−0	=	6·369	6·369 × 5 =	31·845	1=31·845
,, 1	6·369	−2	=	4·369	4·369 × 3 =	13·107	2=44·952
						44·952	
,, 2	4·369	−3	=	1·369	1·369 × 6 =	8·214	3=53·166
						53·166	
,, 3	1·369	−2	=	−0·631	−0·631 × 4 =	− 2·524	4=50·642
						50·642	
,, 4	−0·631	−3	=	−3·631	−3·631 × 5 =	−18·155	5=32·487
						32·487	
,, 5	−3·631	−1	=	−4·631	−4·631 × 7 =	−32·417	
,, F_2	4·631	−0	=	4·631		·070	

In designing a girder, we have first to consider the shearing actions and the bending actions to which it is subjected. The calculation of the shearing force having been made by previous rules, sufficient sectional area of metal must be introduced in the web if a plate girder, or in the diagonals if a lattice or Warren girder, to withstand this action. Again, having calculated the greatest bending moment, sufficient sectional area must be introduced in the top and bottom flanges to withstand this action.

To express this by symbols for a simple case of a wrought-iron plate girder, let S = span of girder, D = depth of girder, both in feet; let W = total load in tons

uniformly distributed; then $\frac{W}{2}$ = greatest shearing action in tons, and $D \times 12$ = depth in inches. Let t = thickness of plate of web in inches, and s = safe shearing strain of material in tons per square inch, then

$$D \times 12 \times t \times s = \frac{W}{2}, \text{ and } \therefore t'' = \frac{W}{2 \times D \times 12 \times s}.$$

Let s_1 = safe tension or compression to which the metal in the flanges may be subjected, and therefore in this case may be taken, at least for short spans, as about equal; then greatest bending moment $= \frac{W \times S}{8}$ and $\frac{W \times S}{D \times 8}$ = total thrust or tension in tons through the flanges. If b = breadth of flange in inches, and d = its thickness, also in inches, then $b \times d \times s_1 = \frac{W \times S}{D \times 8}$, and therefore $b \times d$ or area $= \frac{W \times S}{D \times 8 \times s_1}$.

This formula, $M = \frac{W \times S}{D \times 8}$, is common to various forms of girders.

For a plate, lattice, Warren, or tubular girder,

$$M = \frac{W \times S}{D \times 8}.$$

For an arch rib, $M = \frac{W \times S}{R \times 8}$, where R = *rise* of arch.

For a suspension bridge, $M = \frac{W \times S}{D \times 8}$, where D = depression at centre of span.

The depth D, rise R, and depression D, should be calculated from the centres of resistance, or gravity, of the parts to which they belong.

When a rectangular beam is subjected to transverse

strain, fibres which were originally straight become curved; this may be well observed in testing a piece of iron, a certain amount of curvature being produced during the deflection of the bar by the load brought upon it.

In Fig. 20 the fibres are shown by the full and by the dotted lines, the latter representing the condition when a load is applied. This action evidently produces compression in the upper part and tension in the lower, because the originally straight layers have now become bent, and form concentric arcs, of which the nearer are shorter than the more remote from the centre of curvature.

FIG. 20.

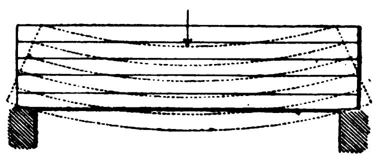
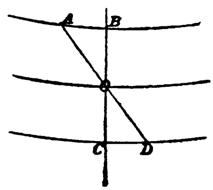


FIG. 21.



These strains may be assumed to vary uniformly from the central layer—which remains unaltered in length, and thus is a neutral or undisturbed layer,—and therefore such a diagram as Fig. 21 will represent this distribution of strain, the area of the equal triangles O A B and O C D representing the total compressive strain and the total tensile strain respectively.

To express this strain by a formula, let A B or C D be expressed by s_1 = strain to which the exterior layer is subjected; let d = depth of beam or B C, and let the breadth of the beam be equal to b; then total compressive strain $= \frac{s_1}{2} \times \frac{d}{2} \times b$. Multiply this by its

leverage, or distance of centre of gravity of O A B from O, $= \frac{d}{2} \times \frac{2}{3}$, and we have $\frac{s_1}{2} \times \frac{d}{2} \times b \times \frac{d}{2} \times \frac{2}{3} = \frac{s_1 d^2 b}{12}$. Again, multiply this by 2, as we have a similar moment in lower or tensile part of beam, and we have

$$\frac{s_1 d^2 b}{12} \times 2 = \frac{s_1 d^2 b}{6},$$

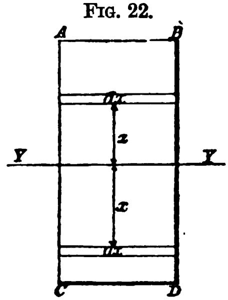
which is the *Moment of Resistance* of the section, and this must be equal to the bending moment, or

$$\frac{W \times S}{8} = \frac{s_1 d^2 b}{6}.$$

The strength of cross sections is sometimes expressed in terms of their moments of inertia, and this arises from the following considerations:—

Let A B C D in Fig. 22 represent a cross section of a rectangular beam, and let Y Y be the neutral axis of the section. Let dx represent the thickness of any layer of a breadth b, and at the distance x from neutral axis; then $b x dx$ will be the moment of that layer. Now, since the resistance to bending of each layer increases uniformly with its distance from the neutral axis, the stress will have the following proportion, $s_1 : s :: x_1 : x$, or $s_1 x = s x_1$, and $\therefore s = \frac{s_1 x}{x_1}$ ($s =$ stress at distance x, and $s_1 =$ stress at distance x_1; x_1 in this example is $=$ half of depth of beam, or $\frac{d}{2}$, because the neutral axis passes through the centre of section). Multiply the moment $b x dx$ by the value of s, and we

FIG. 22.



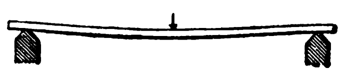
have $\frac{b\,s_1}{x_1}\,x^2\,d\,x$; and taking the integral or sum of these, or $\frac{b\,s_1}{x_1}\int_0^1 x^2\,d\,x = \frac{b\,s_1\,x_1^3}{x_1\,3}$, we get the moment of resistance of the section.

The part of the above integral represented by $b\int x^2\,d\,x$ is called the moment of inertia of the section. The expression $\frac{b\,s_1\,x_1^3}{x_1\,3}$ may be shown to be similar to that already obtained by a different process, for the summation or integration being effected, x has now become x_1; so that we have $\frac{b\,s_1\,x_1^2}{3}$. Again, since $x_1 = \frac{d}{2}$, we have $\frac{b\,s_1\,d^2}{12}$ as the moment of resistance of the upper half of the section, and therefore $\frac{b\,s_1\,d^2}{6}$ = moment of resistance of whole section.

The moment of inertia of cross sections is thus frequently met with in books on mechanics expressed as I, and the moment of resistance as $\frac{s_1\,\mathrm{I}}{x_1}$; so that, as appears from the foregoing investigations, in order to obtain the moment of resistance of a cross section, we take the moment of inertia and multiply it by the greatest stress s_1 to which the material is supposed subjected, and divide by the distance x_1 at which the most severe stress acts from the neutral axis.

Where the load is applied at the centre, the bar does not bend exactly in a curved form, the actual form being more, as shown in Fig. 23. s_1 may be either the ultimate or breaking strength of the material, the working strength, or the proof strength.

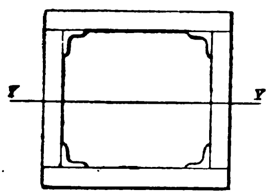
Fig. 23.



Experiments have been made with bars composed of a transparent material, such as glass, whereby, by means of polarized light, the position of the neutral axis was determined.

The moment of resistance of a hollow or of a compound section may be determined by finding the moment of inertia of the cross section, and multiplying this moment by the ratio of the greatest stress to the distance from the neutral axis at which that stress acts.

FIG. 24.



Let A_1 = area of top, and A_2 = area of bottom of rectangular hollow beam, as shown in Fig. 24. Let Y Y be the neutral axis of the beam. Let h_1 and h_2 represent the distance of the centres of gravity of A_1 and A_2 from Y Y, and let b and d represent the breadth and depth of the sides, which are supposed similar, then

$$\text{Moment of inertia} = I = A_1 \times h_1^2 + A_2 \times h_2^2 + \frac{b\,d^3}{12} \times 2$$

$$\text{Moment of resistance} = \frac{I f}{y} = \left(A_1 \times h_1^2 + A_2 \times h_2^2 + \frac{b\,d^3}{12} \times 2\right)\frac{f}{y}$$

f ($= s_1$) being the stress upon the outside layer, and y the distance of that layer or point at which the stress acts from the neutral axis of the section.

We may also determine the moment of resistance of such a cross section by the method previously explained, see page 57. We first find the moment of resistance for the outer section, and then the moment of resistance for the inner section; and the difference of these is the moment of resistance of the hollow section.

Examples of both Methods.—In a plate box-girder, outside dimensions, breadth 10 inches, depth 20 inches,

thickness of plates throughout = $\frac{1}{2}$ inch, greatest safe working strength of the iron = 4 tons per square inch, then

$$M_1 = \frac{4 \times 10 \times 20^2}{6} = 2666 \cdot 6$$

$$M_2 = \frac{3 \cdot 8 \times 9 \times 19^2}{6} = 2057 \cdot 7,$$

and $M_1 - M_2 = 2666 \cdot 6 - 2057 \cdot 7 = 608 \cdot 9$ inch tons. $3 \cdot 8$ tons = resisting strength of iron at outside edge of inner section, or $20 : 19 :: 4 : 3 \cdot 8$.

To determine the moment of resistance of such a cross section by the method of moments of inertia, we have

$$I = \overbrace{(10 \times \cdot 5)}^{\text{Area.}} \times 9 \cdot 75^2 \times 2 + \frac{\cdot 5 \times 19^3}{12} \times 2 = 1522 \cdot 208,$$

and moment of resistance

$$= \frac{If}{y} = 1522 \cdot 208 \times \frac{4}{10} = 608 \cdot 88 \text{ inch tons.}$$

Factors of Safety.

Factors of safety are the ratios existing between the different forms of strength, thus

$$\frac{\text{Breaking strength}}{\text{Working strength}} = \frac{6}{1} \text{ or } \frac{3}{1}.$$

When the load is suddenly applied, or in the condition called a "live load," then we use the ratio $\frac{6}{1}$; if the load is steady, we use $\frac{3}{1}$. If we then are designing, say an iron girder, where the moving load comes suddenly upon it, we take *one-sixth* of the breaking strength of the material for the value of f. If in the design we consider the

weight of the structure itself, then, in allowing a sufficiency of metal to bear its own weight, we take *one-third* of breaking strength for the value of f.

From experiments made by the author, it appears that the tensile strength of cast iron is to the cross breaking strength as 26 : 1. The sections and sizes of bars being for tensile strength links of 1□″ section; for bars, span 36 inches, breadth 1 inch, depth 1 inch, the mean tensile strength was 20,981 lbs., and the mean transverse strength was 805 lbs.; so that by using the equation already given, $\frac{s_1 b d^2}{6} = \frac{W \times S}{4}$, we have, when W = load at centre of bar in lbs., and S = span in inches,

$$(1.)\quad \frac{s_1 \times 1 \times 1}{6} = \frac{805 \times 36}{4}, \text{ and } \therefore s_1 = 43{,}470 \text{ lbs.},$$

which may be called the extreme rupturing strain of the fibres in lines parallel to the axis of the beam.

(2.) To find the tensile strength of the metal when the cross breaking strength is known, we have

$$\text{T} (= \text{tenacity in lbs. per square inch}) = \frac{\cdot 72 \,\text{W} \times \text{S}}{b d^2}.$$

(3.) $W = \frac{b d^2}{S} \times c$, where W = weight in cwts. at centre of bar, S = span in feet, and c a constant = 22, determined from these experiments.

From the consideration of the distribution of the strains in the cross section of a rectangular beam, it appears that the straining action is most severe at the outside layer, and decreases uniformly until the line of the neutral layer is reached. If, then, the outside layers are under their greatest strain, it is evident that those nearer the centre or neutral layer will receive less strain than what they are

able to bear, so that there is a waste of material in this arrangement. From such considerations it was proposed to adopt a form of an **I** shape, such as is now in use for girders. By this means the outer layers of the rectangular beam are represented by the flanges of the girder; and by making these flanges sufficiently wide, the material which would lie between the outer layer and the neutral axis in the rectangular beam may be transferred or concentrated in the flange of the girder, in which position it offers considerably more resistance to the straining action than in the former case. The flanges, then, of a girder bear the bending action of the load, and the web or part keeping them apart is under the action of shearing or cutting through forces.

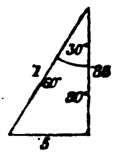
In determining the straining actions in a Warren or zigzag girder we may proceed by two different methods: 1st, by dividing the whole compound frame into a series of simple frames; and 2nd, by considering the action of the forces upon the compound frame as upon a beam.

1. *By the Method of Frames.*

Let the frame 1, 2, 3, 4, be loaded at the points 2, 5, 7, 6, and 3, with loads of 1 ton. Let the angle of the bars be 60°, that is to say, the triangles are equilateral.

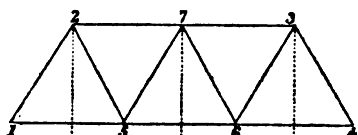
Now let the compound frame be considered in detail, so that the first consideration is to find the strains in the frame 1, 2, 3, 4 (Fig. 25), then we have a load of 1 ton at 2, and a load of 1 ton at 3. Let F_1 be the supporting force, equal and opposite to the downward force at point of support; S_1 = stress on the bar 1, 2, or 3, 4; H_1 = horizontal thrust upon the bar 1, 4. Since the angle

at 1 is an angle of 60°, we have the following relations existing between the forces, $F_1 : S_1 : H_1 :: \cdot 86 : 1 : \cdot 5$, and therefore $\frac{F_1}{\cdot 86} = S_1$ and $\frac{F \cdot 5}{\cdot 86} = H_1$, and if $F_1 = 1$ ton, then we have S_1, or stress on bar 1, 2, or 3, 4 $= \frac{1}{\cdot 86} = 1 \cdot 16$ ton; and H, or horizontal pull on the rod 1, 4 $= \frac{1 \times \cdot 5}{\cdot 86} = \cdot 58$ ton; tabulating these for the first frame, we have



(1.) $F_1 = 1$ ton $=$ load at 2 or at 3.
$S_1 = 1 \cdot 16$ ton on bar 1, 2, or 3, 4.
$H_1 = 0 \cdot 58$ ton on bar 1, 4, or 2, 3.

Fig. 25.



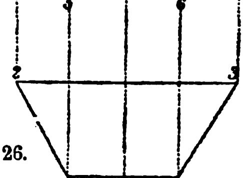


Fig. 26.

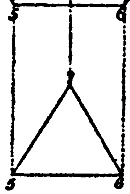


Fig. 27.

We have now to calculate the stresses on the bars of the second frame 5, 2, 3, 6 (Fig. 26); and since we have

also a force of 1 ton at each point of support, or at 2 and 3, we have as the stresses,

(2.) $F_2 = 1$ ton $=$ load at 5 or at 6.
$S_2 = 1\cdot16$ ton on bar 2, 5, or 3, 6.
$H_2 = 0\cdot58$ ton on bar 5, 6, or 2, 3.

Again, in the third and triangular frame (Fig. 27), we have

(3.) $F_3 = 0\cdot5$ ton $=$ load at 7.
$S_3 = 0\cdot58$ ton on bar 7, 5, or 7, 6.
$H_3 = 0\cdot29$ ton on bar 5, 6, or at 7.

The total stresses on the combined frame are therefore as follows:—

Supporting force at 1 or $4 = F_1 + F_2 + F_3 = 1 + 1 + \cdot5 = 2\cdot5$ tons.
Stress along 1, 2, or 3, $4 = S_1 + S_2 + S_3 = 1\cdot16 + 1\cdot16 + 0\cdot58 = 2\cdot9$ tons.
Stress along 1, 5, or 6, $4 = H_1 + H_2 + H_3 = 0\cdot58 + 0\cdot58 + 0\cdot29 = 1\cdot45$ ton.

Supporting force at 2 or $3 = F_2 + F_3 = 1 + 0\cdot5 = 1\cdot5$ ton.
Stress along 2, 5, or 3, $6 = S_2 + S_3 = 1\cdot16 + 0\cdot58 = 1\cdot74$ ton.
Stress along 5, 6, or 2, $3 = (H_1 + H_2 + H_3) + (H_2 + H_3) = 1\cdot45 + \cdot87 = 2\cdot32$.

Supporting force at 5 or $6 = F_3 = 0\cdot5$ ton.
Stress along 7, 5, or 7, $6 = S_3 = 0\cdot58$ ton.
Stress along 5, 6, or at $7 = (H_1 + H_2 + H_3) + (H_2 + H_3) + H_3$
$= 1\cdot45 + \cdot87 + \cdot29 = 2\cdot61$.

2. *By the Method of Sections.*

Total load $=$ 5 tons, half of which is supported at each of the points of support 1 and 4; therefore the vertical forces at 1 and 4 are each equal to $\frac{5}{2} = 2\cdot5$ tons.

Again, the vertical force acting between 2 and 5 is $2\cdot5$ tons $-$ 1 ton $= 1\cdot5$ ton; and finally, the vertical force between 5 and $7 = 1\cdot5 - 1 = \cdot5$ ton.

Tabulating these, we have

$2\cdot5$ tons $=$ supporting forces at 1 or 4.
$1\cdot5$ ton $=$ „ at 2 or 3.
$0\cdot5$ ton $=$ „ at 5 or 6.

And by multiplying these values by the same ratios as in case 1, p. 63, we have for the inclined stresses,

2·90 tons	=	stress along	1, 2, or 3, 4.
1·74 ton	=	„	2, 5, or 3, 6.
0·58 ton	=	„	7, 5, or 7, 6.

The horizontal stresses are found by finding the bending moments at the points 2, 5, 7, 6, and 3 (as explained at p. 54), and dividing the moments so found by the depth of the girder, thus:—

	Tons.
Supporting force at 1 =	2·5
„ „ 2 =	1·5
„ „ 5 =	0·5
„ „ 7 =	0·0

Bending Moments.

Round 2 =	2·5 × ·5	= 1·25
„ 5 =	(1·25 + 1·5 × ·5) ..	= 2·00
„ 7 =	(1·25 + ·75 + ·5 × ·5)	= 2·25

And the horizontal stresses are,

$$\text{At 2 or 3, or their opposites} = \frac{1 \cdot 25}{\cdot 86} = 1 \cdot 45.$$

$$\text{At 5 or 6, or their opposites} = \frac{2 \cdot 00}{\cdot 86} = 2 \cdot 32.$$

$$\text{At 7, or its opposite} = \frac{2 \cdot 25}{\cdot 86} = 2 \cdot 61.$$

On comparison of the two methods it will be seen that the results are similar.

FORMS OF GIRDERS.

Plate Girders.

Fig. 28 shows in section a wrought-iron plate girder; Fig. 29 shows part of same girder in elevation. A and B are the flanges of wrought-iron plates, suitably joined at their extremities by covering plates; C is the web, gene-

rally of thin plate. Angle-irons D are riveted to the web, the flanges being also riveted to the angle-irons. Stiffen-

FIG. 28.

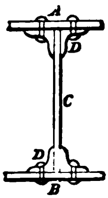
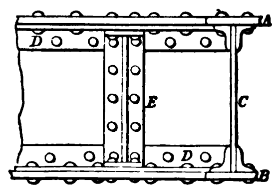


FIG. 29.



ing pieces of tee-iron E are riveted to the web and flanges to give additional stiffness to the structure.

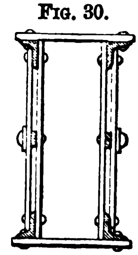
The *pitch*, or distance between the centres of the rivet-holes, is from 3 inches to 4 inches.

When the span is considerable, it is more economical to use a girder having a web of open work instead of solid Of such girders the principal are the

Lattice Girders.

In this form of girder the flanges are separated from each other by a stiff framework of bars arranged diagonally, the angle generally being 45°. These bars at their points of intersection are secured by rivets.

FIG. 30.



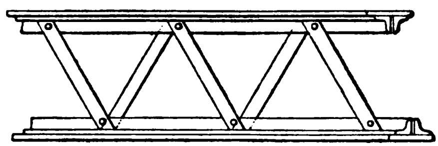
A convenient form of constructing such girders is to have a double web to each set of flanges, the flanges in section being trough-shaped, as shown in Fig. 30.

The number of intersections of the bars will depend on their size and form.

In the Warren or zigzag arrangement of web bars there are no intersections, the bars forming a series of equilateral triangles.

This form of girder has been much used for foreign work, from the facility with which it can be put together,

Fig. 31.



especially when the ends of the struts and ties forming the triangular web are secured by bolts instead of rivets.

The use of rivets is, however, more common now than formerly, as it is found that such fastenings give greater rigidity than bolts.

The Crumlin Viaduct, in South Wales, is a large example of this form of girder. The railway in this case is carried on top. 10 spans, each 150 feet, 14 feet 6 inches deep; top member box form, $14'' \times 9''$; bottom member of flat bars, $6'' \times \frac{5}{8}''$; 18 diagonals in span; struts $+$ section; ties, flat bars.

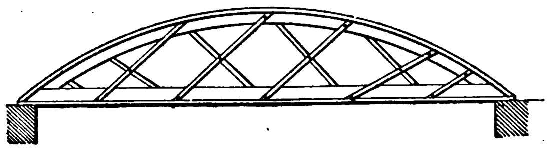
Bowstring Girders.

When the upper flange is curved into a parabolic or circular form, and having its ends secured to the lower flange, the girder is known as the bowstring or bow-and-string girder. In this case the upper flange acts as an arch, its ends or points at which the thrust is exerted being kept in or tied by the lower flange, which thus acts as a "tie." The intermediate space between bow and tie is filled with diagonal bracing, to give rigidity and also to distribute the straining action due to a passing load.

Such girders are suitable for railway traffic, as they are stiff under passing loads.

A convenient form is that having the bow and tie of a trough-shaped form, as exemplified by Figs. 30 and 32; the

FIG. 32.



web bracing being alternately of bars of tee-iron and channel iron riveted at points of intersection. When the span of such girders is considerable, the upper parts of the bow have to be braced together to keep the two girders or frames in place. Triangular framework also serves in a bracket form to give lateral stability. The cross girders rest on the inverted trough formed by the lower member or tie. These girders are covered with planking, along which heavy wooden beams run carrying the rails.

Tubular Girders.

The three great examples of this class of girder are the Britannia Bridge across the Menai Straits in Wales, the Victoria Bridge across the St. Lawrence at Montreal, and the Conway Bridge over the Conway, all by Robert Stephenson. In this form of girder the upper and lower flanges constitute the top and bottom of a rectangular hollow beam; the web, being in duplicate, forms the sides, rails being laid on the lower flanges; the railway traffic passes through this hollow beam. The flanges are made cellular, and, on account of this, are more difficult to keep in good order from the difficulty of repainting.

The Britannia Bridge consists of two continuous girders, each 1487 feet long, resting on three stone towers and two abutments, and forming four spans. Each girder is fixed to the central tower, but is free to move on rollers placed on the other towers and abutments. The two central spans are 459 feet each, and the two shore spans are 230 each. The bridge is 100 feet above the surface of the water. Depth, about $\frac{1}{16}$th span.

The Victoria Bridge is a single continuous girder. Length from bank to bank = 10,384 feet, or about two miles. Centre span, 330 feet; each of the others, 242 feet: 25 spans in all. Width of piers, 16 feet; the two centre piers, 24 feet each. Some of the stones weigh 7 tons (limestone). Depth of centre tube, 22 feet, or about $\frac{1}{16}$th of span. Height above summer level, 60 feet. The central tube weighs about 600 tons, and is lifted at centre $1\frac{5}{8}$ inch by 80° of range of temperature.

The Conway Bridge consists of two tubes of 400 feet span resting on masonry abutments; each tube weighs about 1300 tons. Depth, $\frac{1}{16}$th span.

Such bridges are not now in much favour with engineers, the open or lattice type being preferred.

Arch Ribs.

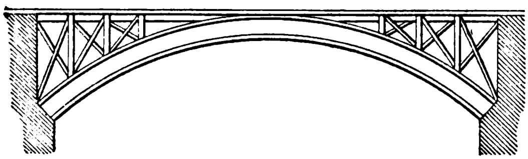
An arch-rib girder may be looked upon as a bowstring girder, whose ends, instead of being kept in position by a tie, are sustained by the reaction of abutments at the springing, as in the case of arches of masonry.

The rib is formed to a parabolic curve, or more generally to an arc of a circle. The form of the rib itself is usually **I** shape, and the depth of the rib at the crown of the arch is usually fixed conventionally as a certain proportion of the span, about $\frac{1}{60}$th, increasing towards the springing.

Along the crown of the rib, and sometimes running into it, is placed a horizontal stringer of plate iron, suitably stiffened by plate and angle irons.

The spandril filling consists usually of a series of verticals at regular intervals and diagonal bars between. By this means the rib and stringer are connected, and loads are transmitted to the rib.

FIG. 33.



Two important matters must be attended to in arch-rib bridges, viz. the proper abutting of the ends of the ribs on the bed-plates, bolted to the masonry, and the proper transverse bracing necessary to bind the two or more parallel frames (made up of the rib and spandril, &c.) together.

In some cases, in order to provide for the expansion of the metal by changes of temperature, the rib is hinged or rests on trunnions at the springing; it seems probable, however, that the friction arising from the great pressure on the joint prevents the desired motion of rotation, so that the rib is probably as well placed when carefully set on bed-plates with plane surfaces.

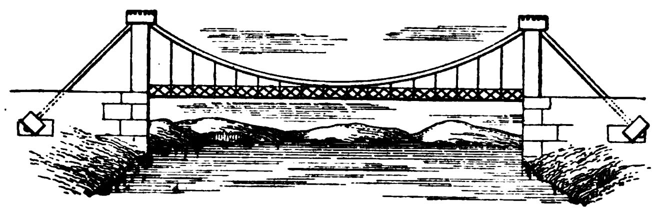
Suspension Bridges.

By means of chains or ropes of wire suspended from supports, roadways may be carried over considerable spaces.

A suspension bridge, then, consists of a chain or wire

rope passing over towers, the ends of the chain being securely fixed or "anchored" in masonry at some distance behind and below the towers. Suspending rods, usually vertical, are placed at regular distances; these support the roadway, which is generally of wooden planking. The chains of the bridge are made of long flat bars connected by steel pins.

FIG. 34.



The great drawback to the more extended use of such bridges is the oscillation of the roadway due to the traffic. Such bridges, however, may be considerably stiffened by a rail or parapet of lattice work.

Any action upon such bridges taking place at regular intervals may seriously endanger their stability, as the tendency is to an increasing series of oscillations.

The Menai Bridge (for road traffic; roadway 30 feet broad; span, 570 feet; height of roadway above high water = 100 feet), by Telford, is a great example of this class of bridge, the chains being formed of long flat bars or links connected by bolts.

The Niagara Bridge (road and railway traffic), span, 1040 feet. Wire ropes in this case act instead of chains to suspend the roadway.

The curve of equilibrium of the chain of a suspension

bridge, when a uniform load rests on the platform, is a parabola; when unloaded, the curve is a catenary.

An example of a combination of bow and suspension bridge, by Brunel, is that at Saltash, in Cornwall. This bridge, erected over the Tamar, consists of two spans of 433 feet; the upper member is an oval-shaped tube 12 feet deep and $16\frac{3}{4}$ feet wide, and stiffened by longitudinal ribs and transverse rings; the ends of the tubes are "tied" by suspension chains; the total depth of combined girder is 56 feet, of this the rise of the bow is 29 feet and the drop of chain = 27 feet.

Greatest range of temperature observed at Saltash = 97°, and the consequent variation = $2\frac{3}{4}$ inches.

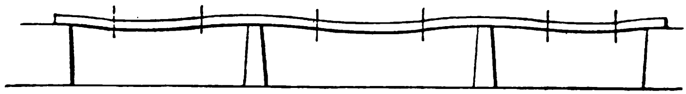
Continuous Girders.

When a horizontal girder is long, there will be so much movement due to the expansion and contraction of the metal from summer to winter temperatures, that some mechanical arrangement must be devised to obviate this action. This is usually accomplished by placing metal rollers in a framework under one end of the girder, the other end being fixed; the free end, therefore, moves backwards and forwards according to the temperature, and thus no prejudicial straining action is set up.

Where a considerable space is to be covered, several spans will be required, these spans resting on suitable supports. In some cases all the spans are connected, thus forming one long girder. Such girders are called *continuous,* and are usually fixed on the middle pier or support, and are thus free to expand on either side. This principle is carried out in the Britannia and some other large structures. The advantage of this arrangement lies in the transference of a part of the bending moment, which

would arise in a free single span, from the centre of span to the point of support or top of pier. This entails

FIG. 35.



changes in the curvature, and consequent change of stress from tension to compression, and *vice versâ*, at and about the points at which these changes of curvature take place. It is evident that, if the various curves which the beam assumes be equal and regular, the points of change of curvature will occur at distances from centre of points of support of one quarter of the distance from centre to centre of two contiguous piers.

When a bridge is constructed across a river-course or estuary having a sandy bed, a convenient method of erecting piers or supports is by sinking cylinders of metal, and by filling the hollow pillar so formed with concrete and masonry.

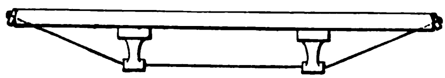
The necessary excavation inside the cylinders is effected in various ways. In some cases by air pressure, and in others by mechanical excavators. Milroy's excavator is a successful application of the latter method to such purposes, the general arrangement being a ring-shaped frame having a series of pointed iron spade-shaped pieces attached in such a manner that, when the whole apparatus is lowered into the cylinders, the sharpened spade-like pieces enter the soil, after which, by means of mechanism connected with the upper surface, these spades are closed so as to draw up and hold the excavated material; the whole is then raised, and the material removed.

COMPOUND BEAMS

(*or wood beams stiffened by an iron tie rod*).

In this form of beam we have the upper or timber portion in compression, and the under or iron part in tension. This is obtained by placing brackets at right

FIG. 36.



angles to the axis of the wooden portion, which hold the iron rod or tie out from the beam, the ends of the tie being secured to the ends of the beam by plates and screws.

The calculation of the strains at central part is similar to that for an ordinary girder, as in this case the wood beam is the upper and the iron rod the lower member; and by finding the bending moment and allowing suitable coefficients of strength, the proper sectional areas may be determined. The strains upon the sloping parts of the tie will be found by the method of triangles.

Let s = horizontal pull on tie rod; then if l = length of sloping part of tie rod, and if d = depth of beam (measured to centres of top and bottom members), the strain upon the sloping part of tie rod will be

$$\frac{s \times l}{\sqrt{l^2 - d^2}} = \frac{s \times l}{h},$$

where $h = \sqrt{l^2 - d^2}$ = horizontal projection of l.

PLATFORMS OF BRIDGES.

In all iron bridges the platform for the roadway is an important element, especially in those for road or street traffic. Where the traffic is heavy, such as in the case of bridges crossing rivers running through large cities, a very strong platform may be made by connecting the horizontal stringer in arch ribs, or the flanges—upper or lower, as the case may be—in the other forms of girders, by cross girders, and covering the rectangular spaces left by Mallet's buckled plates, upon which asphalte, concrete, and granite setts are placed for the traffic.

Where the traffic is light, planking of, say, $10'' \times 4''$ may be laid upon the cross girders (if placed diagonally it will be favourable to rigidity), and upon this flooring cubical wooden blocks of end wood of $4'' \times 4'' \times 4''$ are then laid. The planking should be coated with tar, covered with a thin layer of asphalte, and the blocks set and grouted with hot tar.

The greatest moving load for a road bridge may be taken as 1 cwt. per square foot.

DEFLECTION OF BEAMS.

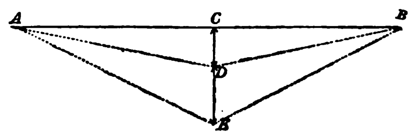
When a beam is under the action of a load, a certain amount of depression or deflection takes place.

If we consider this subject in a general way, it will appear that by increasing the load an increase of deflection will follow; and if we lengthen the span a still greater deflection will be the result.

Again, since by increasing the section of the beam additional strength is given, a decrease in the deflection will ensue. So that generally, deflection $= \dfrac{W \times S^n}{E\, b\, d^n}$.
E = modulus of elasticity of the material.

It must be borne in mind in considering the subject of deflection, that for given increments of extension of a cord or bar supported at its extremities, there will be a more rapid increase of deflection.

FIG. 37.



To give an example: Let A B be a cord or bar which by the action of forces is first brought into the position A D B, and second into that of A E B. Let A B = 20, and C B = 10; and let B D in first case = 10·1, that is to say, C B has been extended by ·1, or $\frac{1}{10}$th of itself; the deflection C D is evidently = $\sqrt{10 \cdot 1^2 - 10^2} = 1 \cdot 41$. Let a further extension of ·1 take place, so that E B = 10·2; the deflection C E is now = $\sqrt{10 \cdot 2^2 - 10^2} = 2 \cdot 01$. The ratio of stretched lengths to original lengths is $\frac{10 \cdot 1}{10}$ and $\frac{10 \cdot 2}{10}$, and if the deflections were proportional to the extensions, we should have

$$101 : 102 :: 1 \cdot 41 : \frac{102 \times 1 \cdot 41}{101} = 1 \cdot 42;$$

but we see from above illustration that the actual deflection in this latter case is 2·01; therefore the deflection increases in a faster ratio than the extension or addition to length of the cord or bar.

This principle is of importance in the chains of suspension bridges.

From experiments made by the author it would appear

that the deflections of similarly shaped beams vary nearly directly as the loads and cubes of the spans, and inversely as the breadths and cubes of the depths, or deflection

$$= \frac{W \times S^3}{b\,d^3} \times c.$$

The mean breaking weight of 771 cast-iron bars of 36 inches span, 2 inches deep, and 1 inch broad, was 3439 lbs., and the mean ultimate deflection was ·377 inch.

The mean breaking weight of 72 cast-iron bars of 36 inches span, 1 inch deep, and 1 inch broad, was 805 lbs., and the mean ultimate deflection was ·626 inch.

To determine the value of the constant c in above formula, we have, when the weight is taken in cwts. and the span in feet,

$$\frac{W \times S^3}{b\,d^3}\,c = \frac{30{\cdot}7 \times 27}{1 \times 8} \times c = {\cdot}377, \text{ and } \therefore c = {\cdot}0036.$$

Ex. 1. $\dfrac{\overset{\text{Cwts.}}{30{\cdot}7} \times 27}{1 \times 8} \times {\cdot}0036 = {\cdot}373''$, actual def. $= {\cdot}377''$.

Ex. 2. $\dfrac{\overset{\text{Cwts.}}{7{\cdot}2} \times 27}{1 \times 1} \times {\cdot}0036 = {\cdot}699''$, actual def. $= {\cdot}626''$.

Ex. 3. $\dfrac{\overset{\text{Cwts.}}{25} \times 27}{1 \times 8} \times {\cdot}0036 = {\cdot}303''$, actual def. $= {\cdot}328''$.

STABILITY OF STRUCTURES.

The object of a retaining wall is to resist the thrust of a mass of earth or sand, or of a body of water.

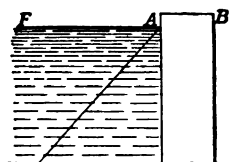
As the latter case is the simpler, it may be considered first.

Retaining Walls to resist Fluid Pressure.

Since the action of fluids is to produce normal pressure on all exposed surfaces, and since the intensity of such

pressure is directly as the depth, we have a ready means of ascertaining the total pressure on any exposed surface, such as the face of a wall of masonry. Let A B C D in Fig. 38 represent in section a wall of masonry which resists the pressure of the water in a reservoir. Let d = depth of water in feet, then, since the weight of a cubic foot of water is 62·4 lbs., $62{\cdot}4 \times d$ = weight of a column of water of a base = one square foot, and of a height = d, and this pressure will be exerted vertically on the bottom of the reservoir. Now, since the pressure of fluids is exerted equally in all directions, there will be a pressure = $62{\cdot}4 \times d$ exerted upon the lowest square foot of surface in the face A C (in such investigations it is usual to consider a breadth of face = unity). Lay off C E = A C; and if A C represent $d \times 62{\cdot}4$, then will C E also represent $d \times 62{\cdot}4$, or the horizontal pressure on the lowest square foot of surface of the wall. It is evident that the pressure on the next higher square foot must be less in proportion to the decreased height, and so on decreasing uniformly to the top at A; the total pressure will therefore be equal to the area of the triangle A E C, so that $\frac{d^2}{2} \times 62{\cdot}4$ = total pressure. The resultant of this pressure acts through the centre of pressure, situated at a distance above C = $\frac{1}{3}$ of A C, and the moment therefore will be $\frac{d^2}{2} \times 62{\cdot}4 \times \frac{1}{3} d = \frac{d^3}{6} \times 62{\cdot}4$.

FIG. 38.

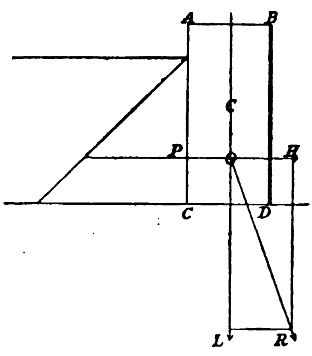


Now, if we wish to ascertain what weight of wall would just withstand the fluid pressure, we must have a balance of moments round the point D.

Let h = height of wall, b = its thickness (the breadth being = unity), and let the weight of the masonry = 120 lbs., then $h \times b \times 120$ = weight, and the moment of this weight $= h \times b \times 120 \times \frac{b}{2} = \frac{h\,b^2\,120}{2}$; $\frac{b}{2}$ is taken as the arm of the force, as the wall in this example being rectangular, the resultant of the weight passes through the centre of figure, which is also the centre of gravity. We have therefore the equation $\frac{d^3}{6} \times 62 \cdot 4 = \frac{h\,b^2 \times 120}{2}$, so that by fixing the depth of water and height of wall, we can determine the thickness with which the wall would just stand, or $b = \sqrt{\frac{d^3 \times 62 \cdot 4}{3 \times h \times 120}}$.

The stability of the wall may be also determined by constructing a diagram of pressures as follows. Calculate the total pressure due to the fluid as before, and to scale, and at a distance C P = one-third depth of water, draw the horizontal line P H, and lay off O H equal to the pressure of the fluid.

Fig. 39.



Through G, which is the centre of gravity of the wall, draw the vertical line G O L, and lay off O L = weight of wall; complete the parallelogram O H R L, and join O R. O R will then represent in magnitude and direction the combined force; and according as that line passes *outside* or *inside* of the point D, will the wall be in

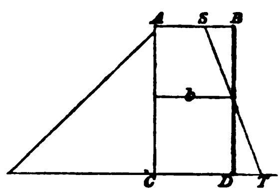
unstable or stable equilibrium. If the line O R pass *through* the point D, indifferent equilibrium will ensue.

It is evident that for safety the line should cut the base C D at some point behind the point D.

In practice the deviation of the point of intersection from the middle of the wall is about one-third of the thickness of the wall at that point.

Such walls are usually constructed with one or both of the faces sloped, or with a "batter," as it is technically termed. Such batter may be varied, but by keeping the middle of the wall of the same thickness as that found by the previous formula, and by decreasing and increasing this by equal parts for top and bottom thicknesses, the weight will still be as before, and the figure will be better adapted to resist the overturning pressure of the water. For example, let A B C D be a rectangular wall as before, and let b = middle thickness, as found by the formula, then if the thickness at top be decreased by S B, and if the thickness at base be increased by the same quantity, or D T, the weight will remain the same.

FIG. 40.



In some large examples of such structures, the faces of the wall are curved surfaces. The curves most suitable are logarithmic. The advantage of such forms lies in the saving of material. In such examples the resultant line of action of the overturning and of the resisting forces is a curve.

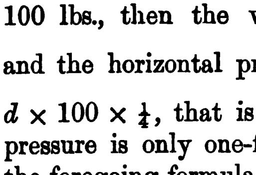
Where earthwork embankments are used for the storage of water, it is usual to make the slope of the inner or water face 3 to 1 (i. e. 3 horizontally to 1 vertically), and that of

the outer face $1\frac{1}{2}$ to 1, or 2 to 1, according to the material employed.

Retaining Walls to resist Earth Pressure.

The same principles apply to earth retaining walls as to those for the storage of water; but in this case the overturning forces due to the horizontal pressure of the earth or other loose material are less than in the case of a fluid such as water. This decrease in pressure is due to the friction of the particles of the earthy material. Various formulæ have been given for calculating the horizontal pressure, that of Professor Rankine being perhaps the simplest. By his method we multiply the vertical pressure (depth $\times$ weight of a cubic foot of material) by the ratio $\dfrac{1 - \sin. a}{1 + \sin. a}$, where a = angle of repose of the material, or the angle which the natural slope of a bank of the material makes with the horizon. Let A B C D in Fig. 41 represent a bank of earth whose natural slope or $\angle$ B A D = 37° (about an average value); then if B D, or d, = depth of earth, of a weight per cubic foot of 100 lbs., then the vertical pressure at D = $d \times 100$, and the horizontal pressure at D = $d \times 100 \times \dfrac{1 - \frac{3}{5}}{1 + \frac{3}{5}} =$ $d \times 100 \times \frac{1}{4}$, that is to say in this case the horizontal pressure is only one-fourth of the vertical pressure. In the foregoing formula $\frac{3}{5}$ is taken as = sin. 37°.

FIG. 41.



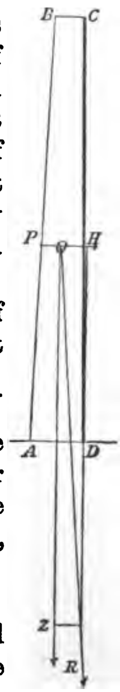
Towers and Chimneys.

Such buildings being exposed to the action of the wind, sufficient stability must be allowed in order to ensure

safety. If the building in elevation be rectangular, then the total pressure due to the wind will be = area × p, where p = wind pressure per square foot; p may be taken as about 55 lbs. per square foot for such structures in this country.

Let A B C D in Fig. 42 represent a tower or chimney under the action of wind pressure on the face B A; let A = area of that face, then A × 55 = total pressure; and if we consider this pressure to be exerted through the centre of pressure, or centre of gravity of the exposed face, there will be an overturning moment equal to $A \times 55 \times H_1$, where H_1 = height of centre of pressure above base. This moment must be resisted by the moment of stability of the structure, which is $W \times \frac{B}{2}$, where W = weight of the building, and B = breadth of base. A balance of moments then will exist when $A \times 55 \times H_1 = W \times \frac{B}{2}$. To ensure stability, the moment of stability must exceed the overturning moment; and by adopting a factor of safety of 2, which appears to obtain in some of the best examples of chimney construction, we have $(A \times 55 \times H_1)\, 2 = W \times \frac{B}{2}$.

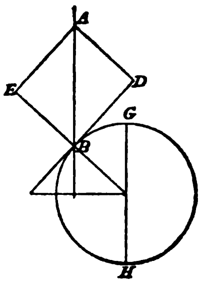
FIG. 42.



In the case of cylindrical or conical shaped chimneys there is a reduction in the effective pressure on the face, on account of the curvature. Let A B in Fig. 43 represent the direction of the wind, which is supposed to be parallel to the diameter G H. Resolve this force, which call p, into two com-

ponents at right angles, A E and A D = D B and E B. D B will exert no pressure upon the face at B, and E B will produce pressure normally at the point B, towards the centre of the figure. E B or A D = p sin. $\angle$ A B D, which, if again resolved as before, will give for effective pressure or force in a line parallel to G H, $p \sin.^2 \angle$ A B D. The mean sine of the arcs affected by the wind will be about ·75, and the square of ·75 = ·56. The mean effective pressure then on the semi-circumference = $p \times$ ·56, so that for a circular chimney we have for total effective pressure of wind A $\times p \times$ ·56, where A = area of a vertical section through a diameter.

FIG. 43.



In the case of a square chimney we have A $\times p$, so that the action of the wind upon a round chimney whose diameter is equal to the side of a square chimney, is as A $\times p \times$ ·56 : A $\times p$, or as ·56 : 1, i.e. about one-half.

Very large examples of chimney construction have been erected in Glasgow at the works of Messrs. Tennant and Coy, and of Mr. Townsend. In the former example the batter is curved, whilst in the latter it is straight.

In designing such buildings, care must be taken to have sufficient area of material in any horizontal cross section to resist the crushing tendency of the material above the section. This may be accomplished by varying the thickness or the diameter, or both.

Professor Rankine, in his 'Manual of Civil Engineering,' states that the deviation of the centre of pressure from the centre of any bed joint should not vary more from the

central line of the building than *one-third* of the diameter of the joint in round chimneys, nor by more than *one-fourth* in square chimneys.

STRENGTH OF SOLID CYLINDERS.*

Shafts and Axles.—When a cylindrical body such as a shaft is subjected to a twisting stress, the axial lines of the concentric layers, of which we may conceive the shaft to be made up, suffer angular displacement, the outer layers being more affected than those nearer the centre, the central layer being unaffected. It is evident from this that if the fibres are free to move while undergoing this strain, a shortening in the axial length must take place. The fibres of the axle are, however, supposed to be fixed to a body such as a crank, and therefore are not free to move; a consequent extension must therefore arise, which practically we may consider as varying uniformly from its greatest intensity at the surface to nothing at the centre.

Let s_1 = greatest strain and r_1 = greatest radius of cylinder, and let s represent the strain at any other radius r; then, since the strain is considered as uniformly varying, $s_1 : s :: r_1 : r$, or $r s_1 = r_1 s$; and $\therefore s = \frac{r s_1}{r_1}$. Let $d r$ represent the breadth of a ring at a distance r, then $2 \pi r d r$ = area of that ring, and $2 \pi r d r \times \frac{r s_1}{r_1}$ = strain on that ring. This, again, must be multiplied by r as leverage, so that $2 \pi r d r \times \frac{r s_1}{r_1} \times r = \frac{2 \pi s_1 r^3 d r}{r_1}$ = moment of resistance of the ring, and $\frac{2 \pi s_1}{r_1} \int r^3 d r =$

* More detailed investigations of the stresses in solid and hollow cylinders were published by the author, in 'Iron,' 1873.

moment of resistance of cross section, or when the integration is effected $= \frac{\pi s_1 r^3}{2}$. If this moment be expressed in terms of the diameter d, then we have

$$\frac{\pi s_1 \left(\frac{d}{2}\right)^3}{2} = \frac{\pi s_1 d^3}{16}.$$

Now, since this moment must equal the moment of the applied forces tending to twist the axle, or $F\,l = \frac{\pi s_1 d^3}{16}$, where F = force applied at leverage l, such as the effort on the crank-pin of an engine, acting at the distance, l = length of crank between centres, we may express the diameter of a shaft in terms of the force exerted, such as in proportion to the horse-power of an engine.

Let indicated horse-power $= H = \frac{p \times A \times 2 S \times N}{33000}$, where p = effective pressure in lbs. on piston per square inch, A = area of piston in inches, S = stroke in feet, and N = number of revolutions per minute; then by substituting F and l in the above, we have, when allowing 5000 lbs. per square inch as safe stress $= s_1$,

$$d'' = \sqrt[3]{\frac{H \times 100}{N}}.$$

The value of d may be expressed in terms of the total pressure on piston and length of crank, or

$$d'' = \sqrt[3]{\frac{p\,A\,l}{80}},$$

l = length of crank in feet.

STRENGTH OF HOLLOW CYLINDERS TO RESIST INTERNAL PRESSURE.

1. *Thin Shells, such as Boilers.*

Rule. $t = \frac{d\,p}{2\,s}$, where t = thickness in inches;

d = diameter of cylinder or boiler in inches;

p = pressure in lbs.;

s = ultimate stress in lbs. per square inch.

Example.—A cylindrical boiler of 7 feet diameter is worked at a pressure of 40 lbs. per square inch; what is the thickness of plate? Here $d = 84''$, $p = 40$, and let $s = 28{,}600$ lbs. per square inch. Let a factor of safety of 6 be taken, then $t = \frac{84 \times 40 \times 6}{2 \times 28600} = \cdot 35$ inch, or about $\frac{3}{8}$-inch plate—a usual thickness.

2. *Thick Hollow Cylinders, such as Hydraulic Cylinders.*

When the shell of a hollow cylindrical body is of considerable thickness, the distribution of the stress due to internal pressure is no longer uniform, but is greater at and near the inner surface, and decreases towards the exterior surface.

The principles of variation of this stress have been investigated by various writers, and rules published whereby the proper thickness of metal may be calculated to withstand a given pressure. From an investigation by the author on this subject, the following rule was determined:—

$$\frac{d\,p}{2} = s_1 \times \text{hyp. log.}\ \frac{R}{r} \times r,$$

where

d = internal diameter of cylinder in inches;
p = pressure in lbs. per square inch;
s_1 = safe *working stress* of material of cylinder per square inch;
R = exterior radius; and
r = inner radius of cylinder in inches.

Example.—Let $d = 8$ inches, $p = 2$ tons, or 4480 lbs.; let $s_1 = 6000$ lbs.; then

$$\frac{8 \times 4480}{2} = 6000 \times \text{hyp. log.} \frac{R}{r} \times 4,$$

$$\text{or} \quad \frac{8 \times 4480}{2 \times 6000 \times 4} = \text{hyp. log.} \frac{R}{r},$$

and $\therefore \cdot 746 = \text{hyp. log.} \frac{R}{r}$.

Referring to a table of hyp. logs., we find that ·746 represents the number 2·11; the ratio then of R to r is as 2·11 is to 1, or $\frac{R}{r} = \frac{2 \cdot 11}{1}$.

Since $r = 4$ inches in example, then $\frac{R}{4} = \frac{2 \cdot 11}{1}$, and $\therefore R'' = 8 \cdot 44$, and $\therefore R - r$ or $t = 8 \cdot 44 - 4 = 4 \cdot 44'' =$ thickness of metal required in shell.

It appears that at high pressures the water permeates the metal to a certain extent, thus reducing the effective thickness.

STRENGTH OF SPRINGS.

A spring being a bar of metal in a coiled form, when weights are applied either to compress or extend the coil, we have a corresponding compression or extension of the metal, and therefore this change of figure will be directly as the weight or force W applied, and directly as the

number of coils. Let N = number of coils, D = mean diameter of coil, and d = side of wire if square and diameter if round (round steel is usually preferred by engineers, as the square form is apt to crack at the edges during coiling); then elongation or compression $= W \times N$. If we now vary the diameter, and consider similar parts of such different sized coils as *beams* undergoing bending, we may apply the formula for the deflection of beams already given (see p. 76). We shall thus have the elongation or compression *directly* as $W \times N \times D^3$.

Again, if the section of metal be varied, and if we still consider part of the spring as a beam, we have the deflection or change of curve as $b \times d^3$, or in this case, since $b = d$ as d^4, and therefore the elongation or compression of the coil will be *inversely* as d^4, and the formula will therefore be

$$\text{Elongation or compression} = \frac{W \times N \times D^3}{d^4} \times C;$$

C = a constant determined by experiment.

If the diameter and thickness of wire be expressed in inches, and the weight in lbs., then it appears from experiment that for steel springs of square section,

$$C = \frac{1}{2{,}200{,}000};$$

and for round section,

$$C = \frac{1}{1{,}470{,}000};$$

the elongation or compression being obtained in inches.

THERMODYNAMICS.

The term heat, as also its opposite cold, expresses certain sensations. Heat and cold may be considered as relative terms, so that when one body is said to be hot and another cold, we simply mean that there is *less* heat in the one than the other. The sensation, then, of heat and cold is, to a certain extent, a measure of the intensity of the heat.

In order to have a reliable heat indicator, such instruments as thermometers have been made, in which a liquid changes its volume through the influence of heat; this change of volume being measured by a scale graduated from a fixed point.

Such an instrument, then, registers or indicates the *temperature* of the bodies which influence it.

When bodies are at equal temperatures, there is no tendency to an interchange of heat.

If a body at a high temperature be placed in proximity to one at a low temperature, an interchange of heat takes place, which, when equilibrium is attained, results in an equal temperature of the two bodies. Such interchange of heat is due to *radiation*, and the transmitted heat is called *radiant heat.*

Solar heat is radiated from the sun through space, and was at one time supposed to consist of a substance called *caloric*, and which bodies capable of being burnt were supposed to emit.

It was argued by some that if such were the case, motion might be induced in a finely suspended body, by the striking of the particles of caloric upon it. Such an experiment was tried, but no motion could be detected.

The prevailing hypothesis is now, however, that heat, like light, is the effect of an impulse upon an imponderable fluid or *ether*, supposed to pervade space, this effect being transmitted by an undulatory movement of the ether. Heat, therefore, is due to motion.

SOURCES OF HEAT.

1. *Solar Heat.* — Various theories are set forth as regards the sun's heat, some supposing that it is due to radiation from solid heated bodies, the lost heat being made up to a certain extent by the fall of meteoric matter. One of the latest theories, which is based on the result of spectroscopic observations of the solar phenomena (for the principles of spectrum analysis, see end of book), is that the sun is largely if not wholly gaseous, hydrogen gas being predominant. The internal part of this gaseous mass is not in combustion on account of the great pressure to which it is subjected; the part beyond this is the *photosphere*, where combustion goes on with great rapidity. Surrounding the photosphere is the *chromosphere*, which is composed of the products of combustion. Great convulsions take place, due to the products of combustion after cooling returning to the central parts; and after again being heated, fresh combustion arises, giving rise to the so-called sun spots.

2. *Mechanical Work*, such as by percussion or by friction. When a piece of iron is hammered it acquires heat, and when the motion of a moving body is retarded by frictional resistance, such as a brake, heat is evolved.

From experiment it has been ascertained that the temperature of water may be raised by a mechanical action exerted upon it, such as by a small paddle-wheel;

and the amount of such mechanical work necessary to be spent to raise 1 lb. of pure water at 39° F. through one degree of temperature, has been fixed at 772 foot pounds. The term 772 is known as the *mechanical equivalent of heat.*

3. *Chemical Action*, as in combustion, where the union of the oxygen of the air with carbon or other combining substance produces heat.

4. *Electric Action*, where in the case of a bad conductor of electricity the temperature is raised.

TRANSMISSION OF HEAT.

Heat is transmitted by radiation, conduction, and convection.

1. *Radiation.*—Heat rays are sent out in all directions from every heated body, the intensity of such rays being inversely as the square of the distance from the point of emission.

2. *Conduction.*—In this manner heat is transmitted through solids: those solids which transmit heat more readily are called good conductors, whilst those which transmit it less readily are called bad conductors. A non-conductor would be a body having no power of transmission.

From experiments it appears that silver is the best conductor of heat, and bismuth the worst; the metals copper, gold, iron, and lead occupying an intermediate position, and in the order indicated.

3. *Convection.*—Heat is transmitted through fluids and gases by an interchange of the particles of these bodies, and is a result of specific gravity.

If a vessel containing water be heated at the lower part, the water adjoining that part will receive an addition of heat, and an increase of bulk will follow; and on account of the specific gravity of this part of the water being lowered, an upward tendency will result, from the upward pressure of the colder and more dense water, which will now descend to be in its turn heated. A series of currents are therefore set up in the water contained in the vessel, by means of which the temperature of the water will be raised: this principle is called convection.

Great examples of this form of transmission of heat exist in nature, currents in the ocean and in the air being due to this cause. The Gulf Stream is due to the difference of temperature between the water of the ocean in tropical regions and the water of the polar regions; the water of the tropics being increased in bulk, tends to flow off in a polar direction. By the resultant action of the trade winds, this water is impelled in a westerly direction, and enters the Gulf of Mexico, from whence it issues as a warm current, quite distinguishable from the surrounding waters. As it goes northwards it divides, one portion being deflected towards our own and more northern coasts, whilst the other tends southwards, and is ultimately lost in the tropics.

Polar currents of cold and more dense water set towards the equator, such currents being the originally heated water of the tropics again returning after having been cooled in the northern regions.

From observations made by the 'Challenger' Expedition, it appears that at the equator the surface water has a temperature of 78°, while at about 600 feet below the

surface the temperature is 55°; at greater depths the temperature falls to nearly 32°.

This seems to indicate a supply of cold water from the poles; as from observations made in the Mediterranean the temperature at 2000 fathoms appears to be the same as at 100 fathoms or 600 feet below the surface, the temperature being about 55°.

The trade winds are also due to convection. The heated surfaces of the tropical regions, by acting on the layers of air at their surfaces, induce upward currents of heated air, which ultimately flow off in a polar direction, indraughts of cold air taking place.

The land and sea breezes common in tropical districts are also due to convection. During the day the land rises faster in temperature than the sea, a heated current of air rises from the land, and is followed by an indraught from the colder air above the sea. At night the land parts with its heat faster than the sea, and consequently the air above the land being cooled more than that over the sea, an outflow takes place.

The draught in a chimney is also an illustration of the principle of convection, as the column of air and gases in the chimney being at a higher temperature than an equal column of air outside the chimney, the hot air column is pressed upwards by the cold air column.

EXPANSION.

When bodies are heated, a certain amount of expansion or increase in volume takes place. This is true for solids, liquids, and gases.

1. *Solids.*—The expansibility of metals is of importance in the designing of iron structures.

Iron.—Cast iron appears to expand about $\frac{1}{900}$ part of its length in a range of 180° F.

From observations made at Saltash (see p. 73), it appears that the expansion of large girders is ·00053 of their length for the range of temperature in this country.

A difference of opinion still exists as to the effect of frost or reduced temperature upon the strength of iron or steel.

Some peculiar phenomena are observable in connection with cast iron at various temperatures, the solid iron being seen to float in the molten iron.

With a view to determine the nature of these phenomena, the author made a number of experiments by placing cold metal on the surface of molten metal (cast iron). In some cases the pieces did not sink, but floated upon the surface; in others the pieces sank, but afterwards rose to the surface.

The following conclusions have been drawn from these experiments.

When the cold metal is put into the liquid metal it sinks, but owing to the great heat of the molten mass, it is rapidly heated; and when the consequent expansion reaches a point such that the bulk of the solid metal is a little greater than its bulk in the liquid state, it rises to the surface, and appears there floating, but now so much raised in temperature that in some cases it has the same or nearly the same colour as the liquid metal.

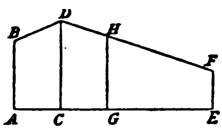
The sinking of the cold metal is only observable when the piece is bulky: if the piece be small, it becomes so rapidly heated that it remains floating.

To carry out such experiments properly, large weighty masses of a spherical form, having great bulk in proportion to surface, should be used: the phenomena being

apt to be masked by the use of small irregularly-shaped pieces.

To illustrate such phenomena, let the vertical ordinates in the diagram represent the bulks of a given weight of cast iron; in the liquid state as A B, in the solidified state as C D, and in the finally cooled state as E F. Join B D and D F, and draw G H parallel and equal to A B. Now, if E F represents the bulk of the cold iron on being put into the molten iron, other ordinates drawn between that ordinate and C D will represent the corresponding bulks of the solid metal, and A B will represent its bulk in the molten state. It is evident, therefore, that until the metal has attained a bulk equal to or greater than what it is in the molten state, represented by G H, it cannot float; and on passing that bulk it will rise to the surface, where it will remain until reaching the bulk represented by C D, when it will pass at once into the molten state, and assume the bulk represented by A B, and thus mix with the molten mass around. The solid metal, therefore, floats when its specific gravity is equal to or less than the liquid metal.

FIG. 44.



The expansibility of metals has been used to indicate changes of temperature, as in pyrometers.

2. *Water.*—The expansion of water by heat has been the subject of inquiry with several eminent investigators.

Mr. Alexander Morton, in a paper read before the Institution of Engineers and Shipbuilders in Scotland, session 1871-72, gives the following formula, which closely agrees with registered experiments.

$$V = 1 + \frac{a t + b t^2 + c t^4}{T^2},$$

where V = volume of water, that at maximum density being = 1;

t = temperature measured from that of maximum density, 39·2° F.;

$$\left.\begin{aligned} a &= \cdot 2863 \\ b &= \cdot 5726 \\ c &= \cdot 0000026913 \end{aligned}\right\} \text{constants;}$$

T = absolute temperature, measured from 461·2° F. below zero.

3. *Air.*—It appears from the various experiments made upon the expansion of air, that a given volume of air at 32° F. increases by $\frac{1}{491}$ part of that bulk for each degree of Fahrenheit.

HEAT INDICATORS.

Thermometers.—Of these we have various kinds:—

1. The Mercurial Thermometer, in which the variation in volume of mercury contained in a glass tube indicates, on a graduated scale, degrees of temperature.

2. The Spirit Thermometer, where the mercury is replaced by coloured spirits of wine. This thermometer is not so quick or sensitive in its action as the mercurial thermometer, but can be used at lower temperatures, as it is not subject to freezing.

3. The Air Thermometer is a differential instrument, and is useful for registering high temperatures.

4. Metallic Thermometers are formed by twisting a piece of wire into a helical form, and fixing a pointer or index hand at the lower end, the whole being suspended in such a manner that the pointer is free to turn round a dial plate, by the application of heat causing expansion in the coils.

Pyrometers consist of a bar of metal with a suitable index. Such a bar on being submitted to a source of heat expands, and a register is obtained by means of the index.

These instruments are used for high temperatures, such as furnaces. The air thermometer is, however, to be preferred.

The graduation of the thermometric scale is effected by choosing certain temperatures as standard points, and dividing the intervening space into a series of equal divisions, which are called degrees. The standard points are the boiling point of pure water at ordinary atmospheric pressure, or 14·7 lbs., and the freezing point of water.

In the Fahrenheit thermometer there are 180 divisions between these standard points; in the Centigrade thermometer 100; and in Réaumur's thermometer 80 divisions, so that, in comparing these thermometers, we have

$$\begin{aligned} \text{Fahrenheit} &= 180 \\ \text{Centigrade} &= 100 \\ \text{Réaumur} &= 80, \text{ or } F : C : R :: 9 : 5 : 4 \end{aligned}$$

and therefore $F = \frac{9}{5} C = \frac{9}{4} R$, so that if we have a certain number of degrees Centigrade or Réaumur, and we wish to change them into their equivalent in degrees Fahr., we multiply by the above ratios, and the result gives the required number of degrees. To this there must be added 32°, as the zero point in Fahrenheit's scale is 32° below the point at which water freezes.

Absolute temperature is an assumed temperature, and is reckoned from the absolute zero of temperature, which is

that point at which a given volume of air would be reduced to an infinitely small volume.

Taking the air thermometer as the standard, it is found that a volume of air at constant pressure, and at 32° Fahr., varies about $\frac{1}{491}$ of its volume for each degree, and therefore at 491° below 32° Fahr. it should have a volume which could only be represented by a mathematical point. The absolute zero then for the three different thermometers will be

$$\left.\begin{array}{ll} \text{Fahr.} = & -459^\circ \\ \text{Cent.} = & -273^\circ \\ \text{R.} \quad = & -218^\circ \end{array}\right\} \text{ below the zero point.}$$

From the above considerations, then, it follows that:—

1. The volume of air or gas at constant pressure varies directly as the absolute temperature.

2. The pressure varies directly as the absolute temperature.

The coefficient of expansion of a gas under constant pressure is the increase of volume for one degree at 32° Fahr., or $\frac{1}{491}$, or $\frac{1}{493 \cdot 2}$ as it is sometimes given.

LIQUEFACTION.

On the addition of heat to certain solid substances, they become gradually softened, and finally liquefy or become molten. In such cases a certain quantity of heat disappears during the change from the solid to the liquid condition. If by the further addition of heat this liquid condition be changed into vapour, an additional quantity of heat disappears. The heat so disappearing, or which cannot be detected by any change in the temperature

of the body, is called latent heat. If a piece of ice at 32° Fahr. be subjected to a source of heat, liquefaction takes place, and the resulting water has a temperature of 32°; the temperature, then, has not varied, although a large quantity of heat has been expended in bringing about the liquid condition. From experiment it appears that it takes 142 thermal units to perform this operation.

A thermal unit is the quantity of heat required to raise 1 lb. of pure water at 39° Fahr. (or point of greatest density of water), through 1° Fahr.

If we still further impart heat to the water, until the temperature rises through 180° Fahr., or to 212° on the scale, the water at that point will be changed into vapour or steam, and a further disappearance of heat will take place; and it has been ascertained that, in order to accomplish this, as many as 966 thermal units are required. The latent heat of water at 32°, then, is 142, and the latent heat of steam at 212° is 966.

To change a given quantity of water at 212° into steam of the same temperature, requires about $5\frac{1}{3}$ times more heat than is required to raise the same quantity of water from 32° to 212°, i. e. through 180° of temperature.

SPECIFIC HEAT.

This term is used to denote the quantity of heat which must be imparted to a body, so that its temperature may be raised through one degree. The specific heats of different substances then indicate their capacity for receiving heat. Such specific heats are all referred to a standard. This standard is the specific heat of water at 39° Fahr., and corresponds to the term already

given, viz. thermal unit; so that expressing the specific heat of water by 1, we can express all other specific heats by multiples or subdivisions of 1. Thus the specific heat of iron is expressed by 0·1138, lead = 0·0293, stone and brick = 0·2000, which shows that less heat is required to raise such bodies through 1° of temperature than would be required to raise the same weight of water through 1°; or in other words we may say, taking the case of iron, that to raise water at the standard temperature through 1°, takes about nine times as much heat as to effect a similar elevation in an equal weight of iron.

In the gaseous condition it appears that liquid substances have their specific heats reduced; thus the specific heat of water = 1, whilst the specific heat of steam = 0·480.

The total quantity of heat, expressed in thermal units, required to alter the temperature of a body by a given number of degrees, may be taken as proportional to the product of the weight of the body, its specific heat, and the number of degrees; or total thermal units = weight in lbs. × specific heat × number of degrees.

Various experimental methods of determining specific heats have been employed; one method being to immerse a heated solid body in water, the weights and the temperatures of both being known; e.g.,

Let W = weight of solid, and W_1 = weight of water;
„ T = temperature of solid, and T_1 = temperature of water;
„ t = resulting temperature, or temperature of the combination;

then specific heat of solid $= \dfrac{W_1 (t - T_1)}{W (T - t)}$.

COMBUSTION.

The heat of combustion is due to chemical affinity. The oxygen of the atmosphere combines with the carbon or hydrogen of the fuel, and an evolution of heat takes place. In all cases of combustion, this combination or union of the substances is rapid.

The elementary bodies or simple substances are those which form such combinations. The number of these elementary bodies or elements known to chemists is 65. There are certain proportions in which combining bodies unite.

Oxides are due to a slow combination of an elementary substance and oxygen. Thus one of the oxides of iron is familiarly known by the term *rust.*

During such combinations heat is evolved, and when the combination is sufficiently rapid light is emitted.

When light and heat are evolved, bodies are said to *burn.* Flame is gas rendered white hot by the heat of combustion.

The term incandescence is used to denote luminosity. Bodies which thus burn are termed *combustibles,* e. g. coal, coke, charcoal, wood.

The principal constituents of these combustibles or fuel are carbon and hydrogen.

Carbon is charcoal in a pure state.

Hydrogen combines with oxygen in proportion to form water, which is the result of the combustion of hydrogen.

The combination of oxygen and hydrogen, as in the oxyhydrogen flame, produces intense heat; the light from such a flame is inconsiderable.

In the combustion of carbon there is no flame.

The combining weights of some of the simple substances are as follows:—

		Combining Weight.
Oxygen	O	16
Hydrogen	H	1
Nitrogen	N	14
Carbon	C	12
Sulphur	S	32

Water consists of 2 atoms of hydrogen and 1 atom of oxygen, or by chemical notation, water = H_2O.

Such expressions as H_2O are termed a group of atoms, a molecule, or a chemical compound. The molecular weight of water then $= 2 + 16 = 18$, or the sum of the atomic weights.

By the union of oxygen and carbon we have carbonic acid, expressed by chemical notation as CO_2. The molecular weight of which $= 12 + 2 \times 16 = 44$.

The atmosphere consists principally of the gases oxygen and nitrogen, in the proportion of 4 parts of nitrogen to 1 part of oxygen. These gases, however, are not in chemical union, but in a state of mechanical mixture, and we may express this condition by the symbol $N_4 + O$.

The combining weights of the various elementary substances are made use of as follows:—

When we burn carbon, and thus form carbonic acid gas, or CO_2, we have, as already stated, the molecular weight of the combination = 44. Now, in order to the supply of the necessary quantity of oxygen, atmospheric air is required; so that to form CO_2 by burning carbon in the air, we must have $C\{2(N_4+O)\}$, which, expressed in terms of the combining weights, is $=12\{2(56+16)\}$, and the sum of which is

$$\begin{array}{r l}
12 & = \text{carbon.} \\
\left.\begin{array}{r} 112 \\ 32 \end{array}\right\} & = \text{air.} \\
\hline
156 &
\end{array}$$

It appears, therefore, that the molecular weight of air in the combination = 144.

Let the weight of carbon so burned = 1 lb.; then, since the atomic weight of carbon = 12 and the sum of the atomic weight of air required for its combustion = 144, we have $12 : 144 :: \overset{\text{lb.}}{1} : \overset{\text{lbs.}}{x}$, or $x = \frac{144}{12} = 12$ lbs. of air required.

The imperfect combustion of carbon produces carbonic oxide, which burns with a blue flame, and may be often observed in ordinary coal fires. The chemical symbol of this gas is CO; there is, therefore, only one-half the quantity of oxygen in this combination of what there is in carbonic acid gas; the number of lbs. of air then required, in order to bring about such combustion, will only be one-half of that required for perfect combustion, or 6 lbs.

The heat of combustion (perfect) of 1 lb. of carbon has been stated as = 14,500 thermal units; and if we divide this quantity by 966, or the number of thermal units latent in steam at 212° F., we have $\frac{14500}{966}$ = about 15 lbs. of water evaporated by the combustion of 1 lb. of carbon.

Fuel.—The principal constituent of fuel is carbon. This carbon may exist in a bulky condition, such as in coal or coke, or in a finely-divided state diffused through hydrogen gas. In the former case it is called *fixed carbon*; in the latter *hydro-carbon*. The most prominent hydro-carbons are naphtha and olefiant gas.

Since in complete or perfect combustion the carbon of the fuel is all combined with oxygen, it follows that in incomplete combustion some of the carbon will remain free or uncombined; and if its temperature be insufficient to cause incandescence, it will appear as smoke, which,

when still further lowered in temperature, will be deposited as soot.

Flame is gas rendered incandescent by heat, and the luminosity will depend very much on the quantity of particles of solid matter which exist in the flame. In the case of the flame of the ordinary gas jet, the lower part is bluish, whilst the upper part is white or yellowish white; this is explained by considering that coal gas consists of a mixture of several substances, the principal of which are carbon and hydrogen. The greater luminosity of the outer part of the flame is due to the particles of carbon contained in the gas having become heated to incandescence.

Coke and charcoal are the solid residues of coal and wood left after the volatile products have been driven off by the application of heat. This is effected by heating the coal or wood in close vessels or retorts. It appears that the proportion of weight of coal to the weight of the coke made from it is as 8 : 5. The weight of coke resulting from a given quantity of coal is then = weight of coal $\times \frac{5}{8}$.

Coal.—There are various descriptions of coal, the value of the different kinds for fuel being dependent on the percentage of carbon.

1. Anthracite coal, the Welsh variety of which contains about 94 per cent. of carbon, and burns without smoke.

2. Dry bituminous coal contains about 75 per cent. of carbon, and burns with little smoke.

3. Lignite, the Cologne variety of which contains about 70 per cent. of carbon.

4. Cannel or Gas coal contains about 49 per cent. of carbon.

It appears that in practice 1 lb. of coal consumed evaporates about 9 lbs. of water. Theoretically, it should

evaporate about 13 lbs. of water, that is, taking coal containing about 86 per cent. of carbon, 1 lb. of carbon evaporating 15 lbs. of water.

Peat.—In a dry state peat contains about 60 per cent. of carbon. Arrangements for the preparation of peat on a large scale consist in cutting the turf into pulp, and passing it through openings into trays; it is then shed dried.

Wood contains about 50 per cent. of carbon.

The efficient combustion of fuel is an important matter, as upon this depends the efficiency of furnaces for raising steam. It appears that only from $\frac{1}{9}$th to $\frac{1}{11}$th of the theoretical heat of the fuel is obtained in the best forms of steam engine. In order to this efficiency, sufficient air must be admitted to the fuel, so that a proper quantity of oxygen may be available for combustion. In practice it is usual to denote the efficiency of a furnace by the number of pounds weight of fuel consumed per hour on a square foot of grate surface; this is known as the *rate of combustion.*

Chimney draught may be obtained by means of tall chimneys, whereby a constant ascending column of heated gases is induced through the influence of specific gravity, as explained at p. 94.

Blast or forced draughts are obtained by forcibly expelling the air from a short chimney, and thus inducing a more rapid inflow of air to the furnace to fill up the partial vacuum produced in the chimney. This principle is adopted in locomotive engines, the rate of combustion being about five times that of ordinary boilers.

STEAM BOILERS.

Boilers may be divided into land, marine, and locomotive.

Of the land boiler there are a variety of types:—The old wagon form of Watt, without internal flue, suitable

for low pressures; the Cornish boiler, with internal longitudinal flue and internal furnace; marine boilers, principally tubular; locomotive boilers, tubular.

The furnace must be such that the combustion of the fuel may be properly effected; for this there must be a sufficient supply of air.

The arrangement of the back part of the flue, if it contains the furnace, is also of importance, as in some cases considerable saving of fuel has been effected by building in brickwork with a sloping surface back from the grate bars.

The efficiency of steam boilers depends largely upon the design.

The heat of the furnace must be transmitted to the water in the boiler so as to effect the greatest evaporation possible, through efficient circulation. It appears that the evaporation is considerably greater over those surfaces upon which the flame or hot gases act at right angles to the surface.

Where water tubes are introduced in a boiler, they should be of such a form and so disposed as to enable the steam to rise freely from their surfaces. The tapering form given to the Galloway tubes furthers this movement.

A temperature of about 600° in the heated gases passing up the chimney gives the most efficient draught.

The proportions existing between the different parts of a steam boiler are necessarily very various, as they depend upon the form of boiler and the nature of the work.

The grate surface or area upon which the fuel is consumed must be such that the rate of combustion is sufficient to keep up the required steam for the engine. If I H P = indicated horse-power, and N = number of lbs. of coal which must be consumed per hour to generate

a sufficient quantity of steam to perform the work of one horse; and if R = rate of combustion, then

$$\text{Area of fire grate in square feet} = \frac{\text{I H P} \times \text{N}}{\text{R}}.$$

Taking a marine boiler as an illustration, let I H P = 1500, N = 2, and R = 20, then area of fire grate in square feet $= \frac{1500 \times 2}{20} = 150$. Three-fifths of the nominal horse-power is sometimes used in calculating the area of grate surface in square feet for marine boilers, or nominal horse-power $\times \frac{3}{5}$ = area of grate surface in square feet.

The heating surface of the boiler may be expressed as a multiple of the grate surface, and will therefore be a variable quantity. In marine boilers it may be taken about twenty-eight times the grate surface. The area of the tube surface in tubular boilers being about five-sixths of the whole heating surface.

The thickness of the shell plating of boilers is necessarily very varied. In land boilers it may be taken as from $\frac{3}{8}$ inch to $\frac{5}{8}$ inch for pressures of about 45 lbs.

In marine boilers in some cases the thickness of plates is 1 inch and even more, as the pressures now used in the compound system reach up to 60 and 70 lbs.

The end plates of boilers are made thicker than the cylindrical portions by about 14 per cent.

Relations between Heat and Mechanical Energy.

From experiment it appears that, in order to raise the temperature of one pound of pure water, at 39° Fahr., through one degree of temperature, there is required an expenditure of 772 foot pounds of work.

The term 772 is therefore known as the mechanical equivalent of heat.

Quantities of heat in thermal units may therefore be expressed in foot pounds of work by multiplying a given number of thermal units by 772.

Specific and latent heats may then be expressed in terms of mechanical work, e. g. the specific heat of steam under certain circumstances may be taken as $\cdot 370$, and $\cdot 370 \times 772 = 285 \cdot 6$ foot pounds.

The latent heat of steam at $212° = 966$, and $966 \times 772 = 745{,}752$ foot pounds.

In order to utilize the mechanical work resident in heated fluids, various forms of mechanism or engines have been devised.

In all such engines it is evident that the great aim must be to utilize the largest percentage possible of the total heat at command. If this heat, representing so many foot pounds of mechanical work, could be wholly utilized, there would be perfect efficiency; the term efficiency being used as before, to denote the ratio of useful to total work.

To express this by a simple formula, let T = number of thermal units received by the engine, and let T_1 = number of thermal units unused by the engine after performing work with the supply T, then $T - T_1$ = number of thermal units used by the engine and transformed into mechanical work. The efficiency then of the engine is expressed by $\frac{T - T_1}{T}$ or $\frac{\text{useful work}}{\text{total work}}$.

These relations constitute the principles of Thermodynamics.

STEAM ENGINES.

The efficiency of steam engines in respect of utilizing the heat of combustion is still very low, notwithstanding the great improvements effected of late years. It appears that only from $\frac{1}{8}$th to $\frac{1}{11}$th of mechanical work due to the heat of combustion of the fuel is ultimately utilized in the best steam engines.

From experiments on the evaporation of water in boilers it appears that about 11 lbs. of water can be evaporated by the combustion of 1 lb. of coal, the feed water being at 100° Fahr., about the best working temperature. The quantity of water evaporated is, however, very variable, depending on the coal used and the form of the boiler; probably from 7 to 9 lbs. will be about a fair average.

It is found that the consumption of coal per indicated horse-power per hour varies from 1·7 lb. to as much as 12 lbs.; probably from 2 to 3 lbs. is about the best average performance. Taking 2 lbs. per indicated horse-power per hour, let us see what the efficiency of such an engine is.

The heat of combustion of 1 lb. of carbon = 14,500 thermal units. Let the coal used contain 80 per cent. of carbon, then heat of combustion of 1 lb. of coal = $14{,}500 \times \cdot 8 = 11{,}600$ thermal units, and therefore of 2 lbs. = 23,200. Now if this be consumed per hour, we have $\frac{23200}{60} = 386$ thermal units per minute, and $386 \times 772 = 297{,}992$ foot pounds per minute. Divide by 33,000 = foot pounds per minute of one horse, then $\frac{297{,}992}{33000} = 9 \cdot 03$ horse-power. The total energy then due

to the 2 lbs. of coal consumed is = $9 \cdot 03$ horse-power; but in the example chosen there was only 1 horse-power obtained from the combustion of the 2 lbs. of coal, the efficiency then = $\frac{1}{9}$.

With a new boiler the consumption of coal will be at its lowest, and will rise to a certain extent with the age of the boiler, on account of incrustation.

Glycerine introduced into boilers appears to prevent incrustation, by forming a soluble compound with the salts of lime; and from its high boiling point, 285°, it does not readily pass off with the steam.

This waste of energy in the form of loss of heat is due to various causes; such as imperfect combustion; insufficient heating surface; insufficient steam space, on account of which the steam leaves the boiler containing water mixed with it; loss by radiation, due partly to want of sufficient protection, such as non-conducting material, upon the boiler.

1. *Imperfect Combustion.*—Where imperfect combustion exists smoke is produced. The regulation of the supply of air to the furnace and of the fuel is therefore an important matter. With a view to perfect this, various methods of mechanical stoking have been invented, whereby the fuel is supplied at a uniform rate to the furnace.

Although 12 lbs. of air is the weight required for the complete combustion of 1 lb. of carbon, it appears that from the method in which it is applied more air is required, varying from a half more to double the above. In some experiments it appears that 5 per cent. of the total heat of combustion is lost by imperfect combustion.

2. *Insufficient or Badly-arranged Heating Surface.*—The arrangement and quantity of heating surface is of consi-

derable importance, and has been already referred to under "Steam Boilers."

3. *Insufficient Steam Space.*—The result of this is that the steam is so mixed with the water that the latter is conveyed with the steam to the cylinder, technically termed *priming*. Various proportions of the relative spaces for water and steam are given by engineers; in some cases it is expressed as a multiple of the capacity of the cylinder. In marine practice the steam space is about twenty times that of the cylinder.

4. *Loss by Radiation and Conduction.*—By covering the exposed parts of boilers with some non-conducting material, such as felt, the radiation of the heat is greatly diminished. Various compositions have been invented for this purpose.

From some experiments made, as much as 20 per cent. of the total heat of combustion appears to be lost through the various causes which may be classed under this head. The heated gases passing off from the chimney, say 5 per cent., with a temperature of 355° Fahr.

Indicated and Nominal Horse-power.

The indicated horse-power of a steam engine is the measure of the available energy of the steam in the cylinder, and may be measured by means of an indicator diagram.

The nominal horse-power is now a conventional term, and is generally many times less than the indicated horse-power.

This system of measurement of power dates from the time of Watt, and was equivalent in his time, when steam of about 7 lbs. per square inch was used, to the indicated

horse-power. The great improvements subsequently effected on steam engines cause the indicated to be in some cases considerably greater than the nominal horse-power, three, four, and five times being common.

Various rules are given for computing the nominal horse-power of engines, but there appears still to be much room for improvement in this respect.

Some of the more particular matters to be attended to in order to promote efficiency in the engine are,—

1. Steam-jacketing of the cylinders.

2. High rates of expansion, to obtain which high pressures must be used.

3. To obtain the full effect of the steam the cut-off valves should be as near as possible to the piston.

1. *Steam-jacketing.*—When a steam-engine cylinder is unprotected on its outside radiation takes place, which lowers the temperature of the interior and induces a consequent condensation of the steam supplied to work the engine: by surrounding the cylinder with a casing, supplied with steam from the boiler or with heated air, this condensation is prevented. This extra supply of heat to the steam is also necessary to counteract the loss of heat which disappears in the form of the work effected by the steam in the cylinder. (See p. 115.)

2. *High Rates of Expansion.*—By utilizing the elastic properties of steam greater efficiency is obtained. This is obtained by allowing a portion of steam to enter the cylinder, and then closing the point of admission the piston continues to be driven forward by the expansive action of the contained steam. This expansion being accompanied with a corresponding reduction in pressure, it is evident that the higher the pressure of admission the greater the extent to whieh the expansion may be carried.

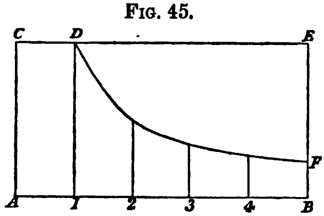
The *rate* or *ratio* of *expansion* is the ratio of the volume of steam at end of expansion to volume at cut off.

If steam be admitted into a cylinder and at one-fifth part of the stroke, or length of travel of the piston during half a revolution, the admission be stopped, the steam will expand, and at end of stroke will occupy a space five times greater than it did at point of cut off. The ratio of expansion in such a case is $5 : 1$ or $\frac{5}{1}$.

The relative efficiencies of expanded and non-expanded steam may be most simply explained by means of a diagram.

Let A B in Fig. 45 represent the length of stroke = S, and let A C represent the initial pressure of the steam $= p$, then if A = area of piston in square inches, we have $A \times p \times S$ = work done when the pressure p is constant, i. e. when there is no cut off.

Fig. 45.



Let the steam be cut off, say at $\frac{1}{5}$th of the stroke, then the pressures at various parts of the stroke after cut off will be inversely as the volumes; and instead of the rectangular figure A C E B representing the work done as in the first case, we have the work now done represented by the figure A C D F B.

The area of this latter figure represents the work done by one-fifth part of the steam. By a comparison therefore of the areas of the figures and of the quantities of steam, we can ascertain the respective efficiencies.

The areas of these figures are in the proportions of $p : p \, \frac{(1 + \text{hyp. log.}\, r)}{r}$, where r = ratio of expansion. This

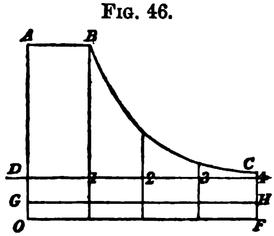
formula refers to steam at a constant temperature, so that any loss of temperature due to condensation or work done must be made good by an extra supply of heat, as by superheating the steam, either by jacketing the cylinder or otherwise.

To give an example:—

	Expansively.		Non-Expansively.			
Cut off.	Volume of Steam.	Proportion of Work done.	Volume of Steam.	Proportion of Work done.	Advantage of Expansion.	Coal used per cent.
0	1	1	1	1	1·00	100
·5	·5	·846	·5	·5	1·69	59
·3	·3	·661	·3	·3	2·10	47
·2	·2	·522	·2	·2	2·61	38
·1	·1	·330	·1	·1	3·30	30

The table of percentage of coal used is calculated by multiplying the reciprocal of the advantage of expansion by coal used without expansion, e. g. if 100 lbs. of coal are used without expansion, then, if the steam be expanded five times, or cut off at ·2 of stroke, the advantage of expansion from above table is 2·61, and the coal now used to do the same work is $= \frac{1}{2\cdot 61} \times 100 = 38$ lbs.

FIG. 46.



The relation between the action of the steam in the non-condensing and condensing forms of engine may be readily understood from a diagram.

1. *Non-condensing Engines.*—O A, Fig. 46, represents the initial *absolute* pressure (or gauge pressure + atmospheric pressure), and F C is the terminal absolute pressure after expansion. O D = F 4 = the

atmospheric pressure (or generally 14·7 lbs. per square inch). D 4 is the atmospheric line. D A will then represent the effective pressure of the steam on entering the cylinder. The ordinate O D will therefore represent the back pressure or resistance to the return stroke of the piston. The whole energy of the steam then for one stroke = area of figure O A B C F O, and the work done in overcoming the resistance due to the back pressure = area of figure O D 4 F O; the effective part of the total energy or useful work done will be equal to the difference of these areas or area of figure D A B C 4 D.

The mean back pressure in non-condensing engines, from various causes is greater than that due to the atmospheric pressure, and is about $16\frac{1}{2}$ lbs. per square inch.

2. *Condensing Engines.*—The diagram for a condensing engine has an additional line G H, which is drawn at some height above the line O F less than O D.

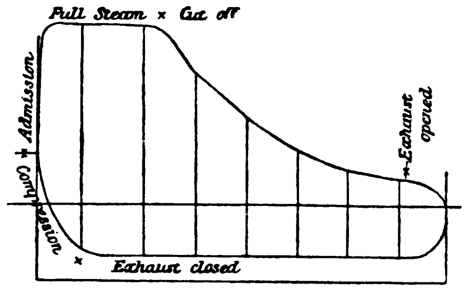
The back pressure in this case is represented by O G = F H, and the work done in overcoming the resistance due to the back pressure = area of figure O G H F O. The useful work done as before is the difference of the areas = area of figure G A B C H G. The advantage gained is evidently = area of figure G D 4 H G.

The mean back pressure, represented in the diagram by O G, is more than that due to difference of atmospheric pressure and vacuum gauge pressure, and may be taken from 3 to 5 lbs. per square inch.

The actual diagram as taken by an indicator is slightly different from the foregoing figures, from various causes due to the action of the steam in the cylinder. The general form of the curves, however, is well preserved. The area of such a figure (see Fig. 47) may be calculated by Simpson's Rule, see page 26: so that if we consider the

ordinates as pressures drawn to scale, and multiply each by its proper multiplier, and multiply the sum of these

FIG. 47.



products by one-third of the common interval (taken as a fraction of unity), the result will be the mean pressure.

Indicator.

Such diagrams are taken by an instrument called an indicator, in which, by means of a small piston balanced by a spring and in communication with the engine cylinder, a tracing pencil point can be moved in a vertical direction.

A rotating drum carrying a slip of paper is placed in such a manner that the pencil connected with the piston of the indicator, when unaffected by the steam pressure, makes a straight horizontal line upon the paper; but when the pencil is moved in a vertical direction by the action of steam upon the indicator piston, a diagonal or curved line is traced upon the paper; this curved line being the resultant of the vertical motion of the pencil and the horizontal motion of the drum. Any variation, then, in the pressure of the steam, such as that due to expansion, is shown upon the paper by a curved line.

COMPOUND ENGINES.

In the compound engine there are generally two cylinders in connection, and so arranged that about half of the work of the steam is done in each. The diameter of the large cylinder is generally about twice or three times that of the small cylinder. The steam, about 65 lbs. per square inch in marine boilers, after being admitted into the small cylinder is cut off about half stroke, and allowed to expand until the piston has completed its stroke; it is then allowed to escape into a receiver, from whence it passes into the large cylinder, where the expansion is completed. The ratio of expansion being as before the proportion borne by the volume of steam after final expansion in large cylinder to the volume in small cylinder at cut off.

Example.—Let the capacities of the two cylinders be in the proportion of 1 to 3, then, if there is no cut off in the small cylinder, the whole expansion being done in the large cylinder, the ratio of expansion will be as 3 is to 1, or $\frac{3}{1} = 3$. If, however, the steam be cut off in the small cylinder during part of stroke, say at $\frac{1}{2}$, then the above ratio is doubled, or ratio of expansion is as 6 is to 1, or $\frac{6}{1} = 6$. In marine practice the ratio of expansion is from 6 to 7.

Such engines have proved very economical. One view of the reason of this is that the high-pressure steam is not under the cooling influence of the condenser, as the large cylinder comes between.

In some cases three cylinders are used, the expansion being carried out in two of the cylinders. Such engines

have been made of such proportions that the diameter of the high-pressure cylinder is to the diameter of each low-pressure cylinder as 73 : 93.

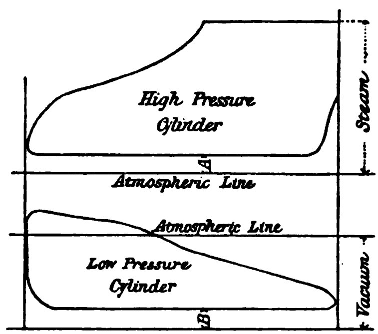
The cranks are usually set at 120°. Some engineers prefer to set the crank of the high-pressure cylinder at 90° with the crank or cranks of the low-pressure cylinders.

Three cylinders may be used with advantage where the power required is great, as the size of casting required for each cylinder is reduced. The motion of the crank shaft will also be steadier and more uniform with the three-cylinder arrangement, a reduction in friction also taking place.

Fig. 48 represents the diagram of a compound engine.

The steam at the end of stroke in the high-pressure cylinder is shown in the figure by A, as being above the

FIG. 48.

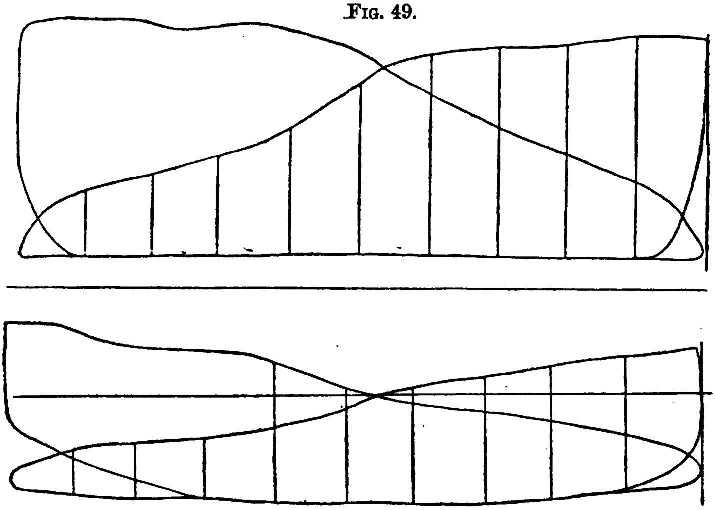


atmospheric pressure, in general a few pounds; this is equivalent to back pressure on the high-pressure piston. The low-pressure diagram shows the pressure at commencement of stroke, as being above the atmospheric pressure; this quantity corresponds, or nearly so, to the

back pressure in the high-pressure cylinder; the expansion is carried out to end of stroke, and as before, the mean pressure of the vapour in the condenser, as determined by the scale of the diagram, such as at B, represents the back pressure in the low-pressure cylinder.

Fig. 49 represents actual diagrams taken from the compound engines of a large screw steamer. Large cylinder, 107 inches diameter; small cylinder, 59 inches diameter.

FIG. 49.



Condensers.

Various forms of condensers have been adopted for forming the vacuum necessary.

The most common form is the air-pump condenser. The exhaust steam from the cylinder is condensed in a vessel by means of a jet of water, and the products of condensation are pumped out by what is called the

air pump into the hot well, from which the boiler is fed by means of a force pump or injector.

The surface condenser is now used on board ocean steam vessels, and consists of a casing containing a great number of small tubes of about $\frac{1}{2}$ inch diameter. Through these tubes a supply of cold water is kept in constant circulation, and the steam coming in contact with their outer surfaces is condensed.

The ejector condenser of Morton is arranged so that the *exhaust* steam passes into a conical-shaped chamber having a trumpet-shaped mouth-piece, where it is condensed by a current of water passing into the above chamber by a conoidal nozzle; this water constantly flows through the chamber. The advantage of this arrangement is that no air pump is necessary, as the energy of the exhaust steam and of the water for condensation is usefully applied in transferring their products to the hot well. In the air-pump condenser this energy is lost, and the pump is therefore required to remove the products of condensation.

McCarter's condenser consists of two chambers, an upper and a lower; the exhaust steam enters the upper chamber, and is condensed by a jet of water.

The lower chamber is in connection with the upper by a valve which is kept shut by a spring. The upper chamber is larger than the lower, and is capable of containing the condensed steam of several strokes of the engine. A vacuum being established in the lower chamber by the admission of steam from the boiler, which is condensed by a jet of water, the weight of water collected in the upper chambers opens the valve, and the water drops into the lower chamber. Air is now admitted to the lower chamber, and the water by its weight opens a valve fitted with a spring, and passes off to the hot well.

HOT-AIR ENGINES.

In this class of engine the pressure due to air when heated is made use of to propel a piston working in a cylinder. There are various varieties of this class of engine named after their inventors. The earliest example is known as Stirling's air engine. This engine has two cylinders, the air in the larger being alternately heated and cooled according to the position of a non-conducting partition or non-air-tight piston which works in it. This is accomplished by heating the lower part of the cylinder and cooling the upper part, so that when its piston is at the lower end of stroke, the air contained in the upper part of cylinder is cooled, and when at the upper part of stroke the air in the lower part is heated. The smaller cylinder is open at the upper part, and has an air-tight piston working in it, from which motion is communicated in the ordinary way to a shaft carrying a fly-wheel. The mechanism is so arranged that the heated air passes from the large cylinder to the smaller, and by its elasticity raises the piston in the latter cylinder. The partition or piston of the large cylinder now returns to the lower part of its stroke, and thus exposes the formerly heated air to the cooling influence of the upper part of this cylinder. The pressure of this air being now reduced, the fly-wheel brings the small piston back again to the lower part of its stroke.

Such engines have not as yet proved so serviceable as steam engines.

GAS ENGINES.

The explosive action of a mixture of gas and air has been used as a motive power.

In Lenoir's gas engine the mixture is exploded in the

cylinder by electricity; the consequent expansion of the heated gases presses forward the piston.

In Hugon's arrangement the mixture is exploded by a jet of gas flame. The cylinders are kept cool by a circulating current of water.

The disadvantage of both these arrangements is the suddenness of the action, and consequent loss of power.

The engine of Otto and Langen appears to have considerable advantage over the other two, as the effect of the explosion is simply used to raise the piston without transferring any of its effects to the machinery. When the piston has reached the top of the cylinder which is open above, the expanded gases have fallen in temperature through the cooling influence of the circulating cold water which surrounds the cylinder, and the atmospheric pressure now acting drives down the piston, whose motion is now connected with the machinery.

The proportion of gas to air used in these engines is from $\frac{1}{9}$th to $\frac{1}{10}$th.

The latter class of engine has been largely adopted for small power, the quantity of gas used being about 20 cubic feet per horse-power per hour.

ELECTRO-MAGNETIC ENGINES.

The motive power in such engines is originally the chemical action in the battery cells where the electricity is evolved.

Various forms of these engines have been made. One of the forms which has been much used for light work is that in which, by means of electro-magnets acting upon ordinary magnets, a reciprocating motion is established, from which rotatory motion is obtained in the usual manner.

It appears that the efficiency of these engines is higher than steam engines, but from the present high price of zinc, which acts as fuel, the working expense is much greater than in the steam engine.

WATER SUPPLY.

Water is conveyed from reservoirs, either natural or artificial, by means of open or close channels; in the open form by aqueducts, and in the close form by pipes. The system of aqueduct conveyance has now largely fallen into disuse, the usual method adopted being that of piping laid underground.

The various parts connected with a water supply are:—

Reservoir, filter, clear-water tank, and pipes. The subject of rainfall and water storage has been already considered (see p. 5), and the stability and proportions of embankments (see p. 81). Where practicable, the main pipe from an earthen embankment of a reservoir should be carried, from its opening inside the reservoir, through the adjoining rock, and not through the embankment itself. When carried through the embankment, it is usual to build a culvert generally of brick, inside which the discharge pipe is laid; this culvert should be well puddled on its outer surface to prevent the passage of water.

The following are some of the best proportions for earth embankments:—

Inner slope, 3 to 1. Outer slope, $2\frac{1}{2}$ to 1.

Height of Reservoir.	Width at Top.	Height above Water Level.
25 feet	6 feet	4 feet.
50 "	12 "	5 "
75 "	18 "	6 "

Least safe thickness of puddle wall at water-line is from 8 to 10 feet, having a batter of 1 inch to the foot or 1 in 12 on each side.

Width of bye-wash at the rate of 3 feet per 100 acres of drainage area.

Shallow reservoirs, or those from 12 to 15 feet deep, are liable to the growth of aquatic plants. This vegetation, however, is lessened by increasing the depth of reservoir.

Great surface, on the other hand, from exposure to the sun's action, favours the colour of the water.

In small works the outlet pipe has a *rose* fitted to its end, which prevents the admission of foreign substances: in large works strainers are used of wire meshes of about sixty strands to the inch; these strainers are arranged in a *well* through which the pipe passes.

The outlet valve is placed on the pipe inside the reservoir, and is generally worked by rods passing upwards through a tower built out from the embankment, reached by a light foot-bridge.

FILTERS.

Filters are large open spaces enclosed by low walls; the bottom is formed of puddle and brick on edge, or paved so as to be water-tight.

The filtering material usually consists of the following materials, arranged in the order indicated.

Sand, 18″ to 2′	2′
Gravel, 12″	1′
Broken whinstone, from 2″ to 3″ in size	2′
	5′

The water being delivered on top.

The whole filtering material rests upon perforated

when put together, on an average of $\frac{3}{8}$ inch of annular space all round; into this space rope yarn is driven and kept in place by lead run in molten and staved hard.

The turned and bored joint is similar in form to the plain or lead joint, but has a pair of belts cast, one in the socket and the other on the spigot; these belts are afterwards turned to gauges, having an inclination of about 1 in 32.

When the pipes are laid, the one is driven hard into the other, the belts, which are usually coated with thin Portland cement or anti-corrosive paint, thus forming a water-tight joint.

Lead joints are required in curved parts of the line of track.

Pipes for water purposes should be cast vertically and in "dry-sand" moulds, and should be "coated" by being immersed in a hot mass of pitch and oil, and kept there until the composition has entered the pores of the metal, so that when the coating is dry it should have a fine glazed surface, which does not rub off.

This process is known as Dr. Angus Smith's. The temperature of the hot composition is about 500° or 600° F.

Valves.—Slide valves are usually adopted, worked by rods and screws.

Air cocks are placed on the higher parts of the line of piping to allow of the escape of air carried inwards by the water.

Scour cocks are placed in the hollow parts of the line to allow of any sediment being drawn off.

Pressure is either calculated in feet of water, measured vertically, or in lbs. per square inch:

$$\text{feet of water} \times \cdot 43 = \text{lbs. per square inch,}$$

$$\text{and } \frac{\text{lbs. per square inch}}{\cdot 43} = \text{feet of water.}$$

FLOW OF WATER THROUGH PIPES.

The flow of water through pipes and channels is affected by the size and condition of the surface of the channel; the smoothness, therefore, of a water pipe or channel reduces the friction.

The flow of water is due to head or difference of level.

If a quantity of water be stored up in a reservoir and an opening be made so as to allow the water to escape as a jet, the theoretical velocity of flow $= v = \sqrt{2gh}$, or $h = \frac{v^2}{2g}$. In calculating the quantity of water which would pass through such an opening (if sharp edged), we must multiply the actual area of orifice by $\frac{5}{8}$, as the jet of water contracts at a short distance from the orifice; this point being known as the "vena contracta." This velocity, however, is much reduced by the friction of the orifice. If we lead this water from the reservoir, by means of a pipe, to some lower point, the head is increased, but the friction has also increased.

It is usual to summarize the various losses to which the velocity of the water is subjected, as loss of head. In the formula for theoretical velocity given above, we have $h = \frac{v^2}{2g}$; h then may be said to be the head lost to give the water the velocity v.

Certain coefficients determined by experiment are used in such calculations.

If water passes through a pipe or channel, a certain resistance is experienced, so that the velocity of discharge is less than the theoretical velocity as calculated from the actual head of water.

The total loss of head, then, is the sum of the losses of head due to velocity and to friction respectively; further losses of head may exist, as at changes in the figure of the pipe or channel.

Generally speaking the friction of water in pipes or channels varies directly as the length of pipe or channel, and as the square of the velocity; and inversely as the diameter of pipe or width of channel; or friction $= \frac{L \times v^2}{D}$. If we express this friction as loss of head, then the loss of head due to friction or $h_f = \frac{L \times \frac{v^2}{2g}}{D}$. By making experiment, a suitable constant can be obtained, or

$$h_f = \frac{L \times \frac{v^2}{2g}}{D} \times c, \text{ and } \therefore c = \frac{h_f \times D}{L \times \frac{v^2}{2g}}.$$

A formula sometimes used for calculating the velocity is $v = 50\sqrt{\frac{D \times H}{L + 50\,D}}$, where v = velocity in feet per second, D = diameter of pipe in feet, L = length of pipe in feet, and H = head in feet.

From experiments made by Mr. J. M. Gale, C.E., of the Glasgow Water Works, and from the calculations of the late Professor Rankine, based on the data obtained, the following coefficient of friction for the flow of water through a 4-foot pipe was obtained, as $f = 0 \cdot 00509$.*

f in this case corresponds to c in above formula, and in determining the value of f, $\frac{D}{4}$ instead of D was used.

* See 'Trans. Inst. Engineers and Shipbuilders in Scotland,' vol. xii.

K

Darcy's formula coefficient is

$$f = 0 \cdot 005 \left(1 + \frac{1}{\text{dia. in inches}}\right);$$

by using this formula for the 4-foot pipe, the obtained coefficient is $f = 0 \cdot 0051$.

Darcy's formula is suitable for all sizes of pipes.

These coefficients are valuable in connection with water supply, as they enable the engineer to calculate the total head required for the discharge or supply required.

SPECTRUM ANALYSIS.

When a beam of solar light is allowed to fall upon some refracting medium, such as a prism of glass, a decomposition of the light takes place; and, instead of the white light which entered the prism, we have, after refraction, a band of various colours.

Beyond the edges of the spectrum are other rays, invisible to the eye, those at the violet end having a chemical power, whilst those at the red end are possessed of heating properties; the coloured rays towards the edges have also these properties.

If the ray or pencil of sunlight be allowed to pass through a narrow slit before entering the prism, it will be found that the spectrum now displayed is marked with a number of fine dark lines called "Fraunhöfer lines," from the observer who first accurately described them. These lines have been carefully studied and laid down to scale so that their relative position may be noted.

If, instead of a ray of sunlight, which is white, or colourless, a ray of light from a coloured flame be allowed to fall upon the prism after having passed through the slit, a spectrum traversed by bright lines will appear, and these bright lines are always different for differently

coloured flames. If, by means of electricity or by the blowpipe, certain substances be heated so as to appear in the form of luminous vapour, each substance can be detected by the particular bright band in the spectrum of its light. A flame containing soda appears to the eye of a yellowish colour, and when its spectrum is viewed *one bright yellow* line is seen.

The purple-coloured flame which appears when potash is present shows in its spectrum *two* bright lines, red and violet. By this means or spectrum analysis chemists are enabled to detect very minute portions of substances, and they have added several new elementary bodies to those previously known.

To perform spectrum analysis with accuracy, complicated apparatus is necessary, but, by a simple arrangement, we can see the bright lines due to certain coloured flames.

If the light from an ordinary gas jet be allowed to fall on a prism, after having passed through a slit cut in a piece of card, it will be found that the band of rainbow colours will be seen when we look at one of the sides of the prism; if a little soda be now added to the flame, a bright yellow line will appear through the yellow-coloured part of the spectrum. By placing a little potash in the flame a red line will be observed, outside and distinct from the visible red of the ordinary spectrum; a violet-coloured line may also be noticed at the extreme edge of the part coloured violet. By burning various substances in this manner the particular lines in their spectra may be observed.

Spectrum analysis has been successfully applied to the Bessemer process, as indicating the exact time at which the carbon is burnt out of the pig iron, by the disappearance of the carbon lines.

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